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Progress Report
for

A RESEARCH PROGRAM TO REDUCE INTERIOR
NOISE IN GENERAL AVIATION AIRPLANES
KU-FRL-317-6
NASA Grant NSG 1301

EXPERIMENTAL AND THEORETICAL
SOUND TRANSMISSION
THROUGH AIRCRAFT PANELS

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Among the many excellent classes the author has attended at the University of Kansas, those of Professors C. T. Lan and E. J. McBride have been outstanding. The author wishes to thank Drs. McBride and Lan for having so enriched his Engineering education, and for serving on his Graduate Committee.

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SYMBOLS

<u>Symbol</u>	<u>Meaning</u>	<u>Usual Dimension</u>
a	Effective Axial length of acoustic cavity	m.
b	Height of acoustic cavity; Edge length of square panel	m.
B	Cavity/Panel Coupling Coefficient	-
C	Width of acoustic cavity	m.
c	Mechanical damping	lb sec/in
c_o	Acoustic velocity	m/s
C_A	Acoustic Capacitance	m^3/N
D	Plate Stiffness $\frac{Eh^3}{12(1-\nu^2)}$	Nm
E	Modulus of Elasticity	N/m^2
f	Frequency	Hz
h	Thickness of plate or panel	m.
j	$\sqrt{-1}$	
k	Mechanical Stiffness	lb/in
K	Variable defined in Eq. 4.14	-
K'	Variable defined in Eq. 4.15	-
l	First acoustic cavity mode number (some- times omitted)	-
m	Second acoustic cavity mode number	-
M	Mass	lb sec ² /in or gram
n	Third acoustic cavity mode number	-
P	Pressure	N/m^2
P	Force (Chapter 6)	N or lb
q	First panel mode number	-

SYMBOLS (continued)

<u>Symbol</u>	<u>Meaning</u>	<u>Usual Dimension</u>
r	Second panel mode number	-
R	Resistive Impedance	
r_A	Acoustic Resistance	$N \text{ sec/m}^3$, i.e. rayl
S	Area of Region of a panel	m^2
s	Laplace variable	sec^{-1}
t	Time	sec
X	Reactive Impedance	rayl
x	Axial location in cavity (See Fig. 4.3.2)	m
y	Vertical coordinate in cavity	m
z	Lateral coordinate in cavity	m
Z	Acoustic impedance	$N \text{ sec/m}^3$, i.e. rayl
α	Acoustic function of time	m^{-1}
δ	Mechanical Damping ($\delta=2\zeta$)	-
ζ	Mechanical Damping Ratio	-
λ	Non-dimensional Natural Frequency	-
ν	Poisson's Ratio	-
π	3.14159...	
ρ	Volume density	kg/m^3
τ	Transmission Coefficient	-
ω	Frequency	rad/sec

SYMBOLS (continued)

Subscripts

Meaning

i	incident
m	cavity mode
n	cavity mode
q	panel mode
r	panel mode
t, trans	transmitted

ACRONYMS

FRL	Flight Research Laboratory
IL	Insertion Loss (acoustic)
KU	University of Kansas
NASA	National Aeronautics and Space Administration
TL	Transmission Loss (acoustic)

CHAPTER 1

INTRODUCTION

This report describes the work performed under a National Aeronautics and Space Administration-funded research project on aircraft-oriented acoustics. This project is intended to provide information leading to the reduction of interior noise in general aviation airplanes.

The more important recent events are presented. This includes a description of project operation, expansions in testing capabilities, and the principal findings since the previous reports (References 1,2, and 3) were published.

Equipment for testing panels at non-normal incidence to sound is a noteworthy addition to the facility. This capability is expected to yield useful data which is otherwise very difficult to obtain, either theoretically or experimentally.

Recent advances made by the project team are expected to very significantly improve the understanding and applicability of the test data which has been and will be obtained. The influence of individual test facilities nearly always "colors" the data acquired. When the phenomena responsible are understood, test data can be applied much more intelligently than otherwise possible.

A compendium of recent test data on panels with holes, on honeycomb panels, and on various panel treatments is also provided.

It is hoped that this report will further the understanding of aircraft acoustics, facilitating the improvements which are so badly needed.

CHAPTER 2

MANAGEMENT AND CURRENT STATUS OF THE PROJECT

This chapter concerns project operation, the project budget, and the schedule of significant project events.

2.1 Historical Update

This section is intended to update the historical account of the project given in Reference 2.

By mid-August of 1977, test facility calibration and testing procedures had been established. At this point, regular testing was begun with bare, treated, stiffened and pressurized metal panels. Much of this data was included in Reference 3, a progress report written in October 1977. Testing continued, and much of the resulting data is contained in Appendix B of the present report.

There were a number of personnel changes in August and September. Mr. C. van Dam, an M.S. candidate at the Delft University of Technology, joined the noise research team. Messrs. T. Henderson and D. Andrews began working at Beech Aircraft Corporation, Wichita, in acoustics. Mr. T. Peschier returned to the Netherlands to work at Fokker-VFW in Amsterdam. Mr. D. Durenberger then became student project manager until December, when Mr. C. van Dam assumed that position. Mr. T.-C. Shu, a KU graduate student in Aerospace Engineering, also worked on the noise research team from August until November 1977, when he graduated and returned to Chinese Air Force duty on Taiwan. In November 1977, three undergraduate research assistants, Messrs. T. Grote, K. Hawley, and P. Meredith, began working part time on the project. The most recent significant personnel change was

the addition of Mr. F. Grosveld, M.S.A.E., Delft, to the research team.

In October, design work on the special (slanted-panel) test sections was essentially completed. Due to prior commitments, the carpentry shop could not begin construction until mid November. All the test sections were delivered in December, and the noise research team immediately began preparing them for use. At the time of this writing, this work is continuing.

A proposal for a follow-on grant was submitted to NASA LaRC in December 1977. The emphasis of this proposal was on exploitation of the capabilities of the existing facility. Specifically, it is planned to test several structural features such as stringer-panel damping and a novel double window design. Comprehensive testing of curved metal and plexiglass panels is also planned. This proposal is presently being reviewed by NASA and appears likely to obtain financial support.

2.2 Project Budget

This section updates the budget outline given in Reference 2, a previous project report. This update will concern primarily the second (panel sound transmission) phase of the project. This phase runs from May 1, 1977, to April 30, 1978. The budget items in Table 2.1 need some amplification.

Funds intended for salaries and wages have been completely expended. This has occurred earlier than planned. This is partly due to higher-than-planned employment in the mid-part of the program. In both cases the temporary high employment was used to expedite completion of critical tasks. Based on the likelihood of a follow-on program, work will continue to progress as planned.

No significant expenditures or credits have occurred to materially

alter the "Electronic equipment," "Test specimens," or "Acoustic materials" costs from Reference 2 levels. It is likely, however, that small expenditures (less than \$100) will soon be necessary for test specimens and acoustic materials. The estimated cost of additional acoustic materials for the special test sections is included in the overall cost of the sections.

Additional "Other expenditures" were primarily for supplies, a steel brace for the brick wall, and the printing of project reports.

The underrun from phase I is also shown in Table 2.1.

2.3 Project Schedule

The project has progressed much as was projected in Reference 2, although it has been found necessary to reschedule several tasks.

Figure 2.2 shows the projected and actual progress.

The testing of panels at normal incidence has continued about two months longer than anticipated, due to several factors. Two new research assistants had to be familiarized with the testing procedure, and this generally required several days. Also student assistants could only work part time during the semester, from August to December. Other factors which extended the testing period were the large number of test panels furnished by Beech Aircraft Corporation, and a more time consuming testing procedure. The more lengthy procedure was found necessary for panels which absorbed sound energy; there were many such panels. It is felt, however, that this time was well spent, for it permitted assembling a large, comprehensive data base including many different types of panels.

The October progress report was prepared next. This reduced the time available for completion of design work on the special test sections. The carpenters, it was found, also had a backlog of work to clean up, at

Table 2.1 Noise Research Project Expenditures

	<u>Projected</u>	<u>Actual (12-31-77)</u>
Salaries and wages (including fringe benefits and overhead	42,009	42,348
Electronic equipment	22,115	19,388
Test specimens	1,000	25
Acoustic materials	1,000	114
Construction, Planewave tube	1,500	804
Other expenditures	<u>2,500</u>	<u>887</u>
	70,124	63,566
Underrun, phase I	<u>591</u>	
Total Funds available	70,715	
Subsequent expenditures <u>to 1-31-78:</u>		
Wages & Sal. & F. B. & O. H.		3,732 (Est.)
Special Test Sections		<u>3,337</u>
		7,069
Total Expenditures and Encumbrances to 1-31-78		70,635

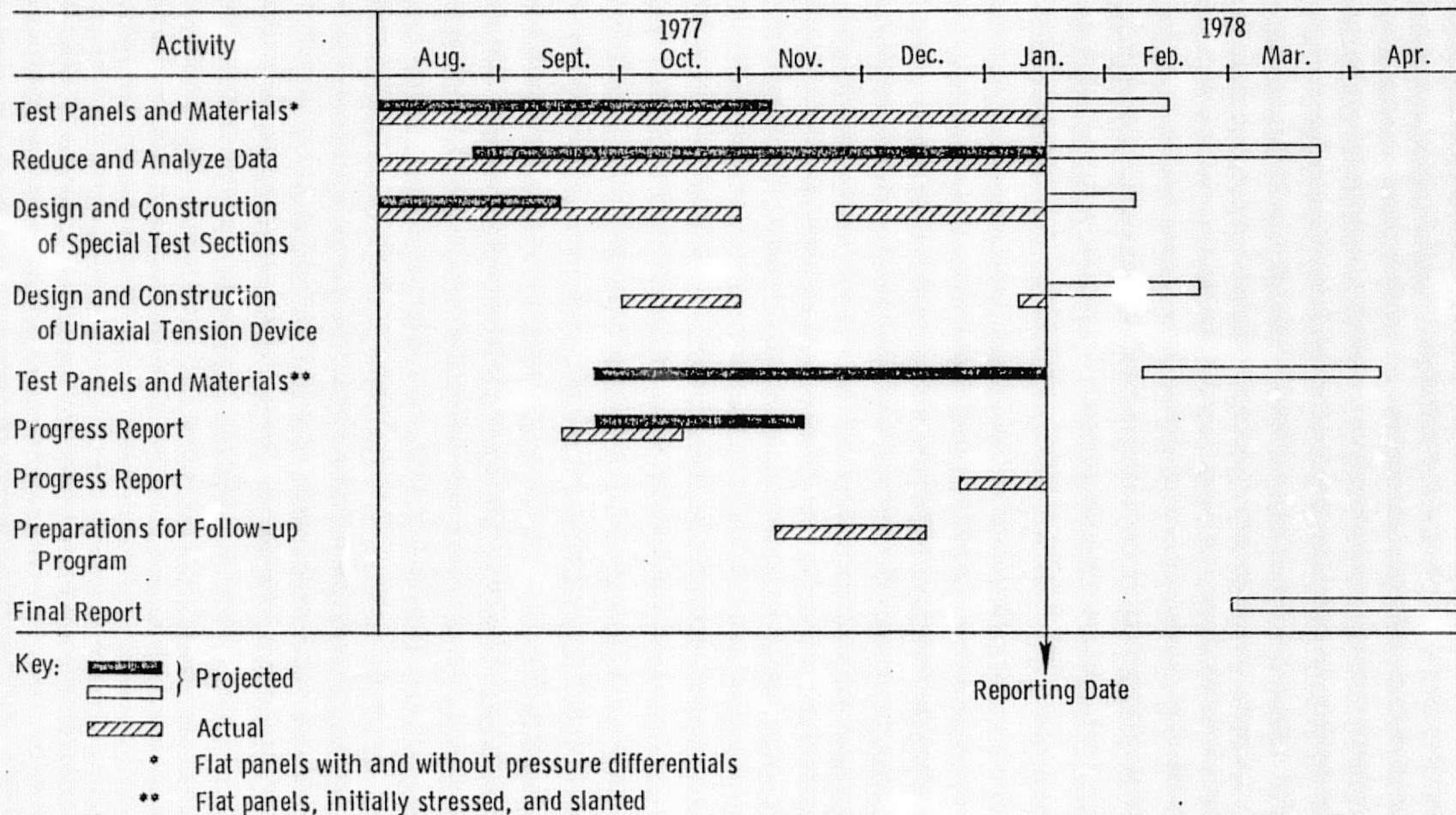


Figure 2.2: Projected and Actual Progress

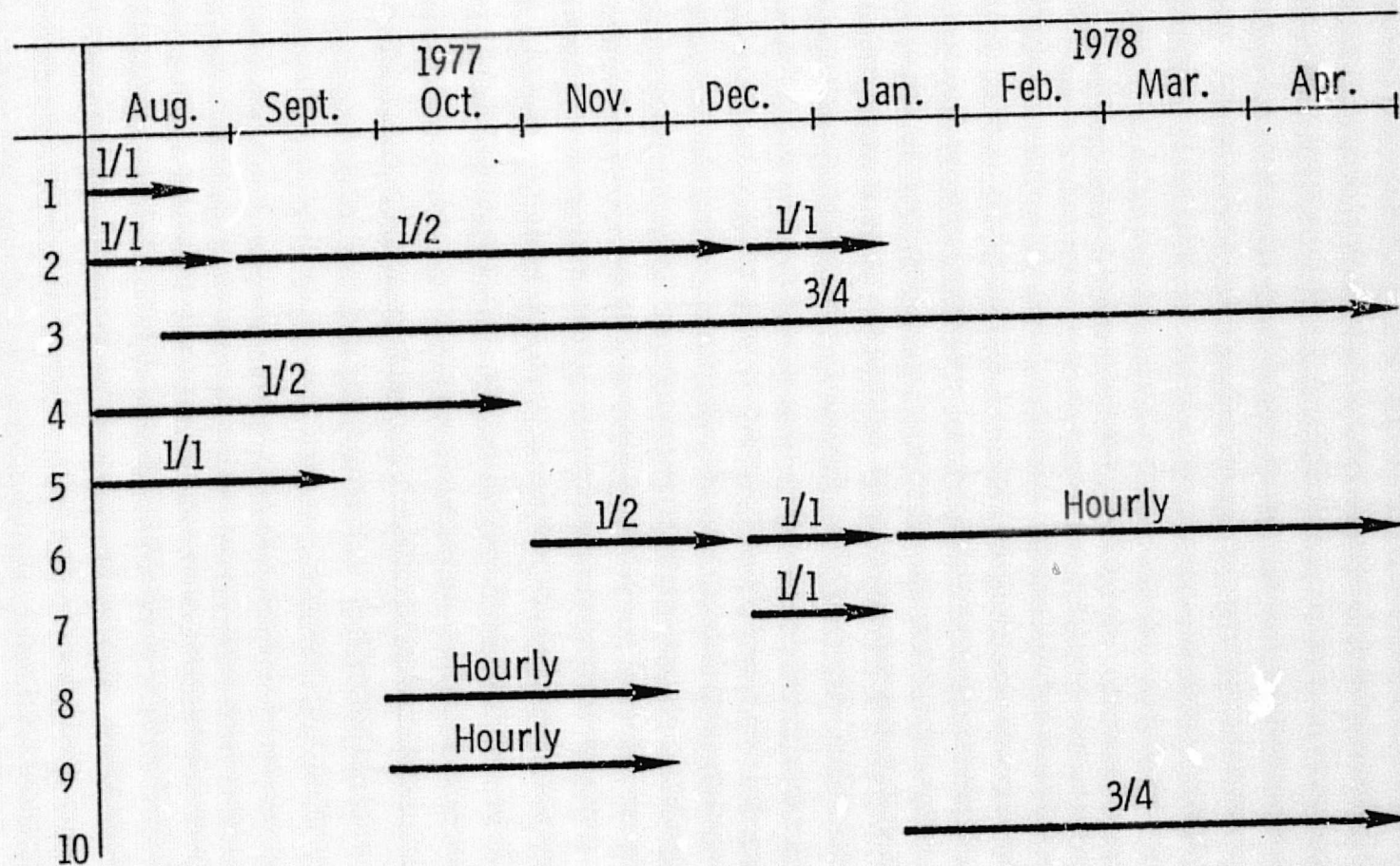


Figure 2.3: Employment of Research Assistants

the time the plans for the special test sections were submitted. Also, at the time Reference 2 was written, the lack of detailed special section drawings caused low estimates of construction time and of time necessary to prepare the sections for use. The necessary preparations included fabrication and installation of clamping and connecting hardware, as well as careful installation of sound absorbing material inside the walls of the special test sections. This work has consumed more time than expected, but is close to completion at the present time. Calibration and testing are expected to begin in late February.

A proposal for a follow-on program was completed in mid-December. This was about four months ahead of schedule. The writing also took less time than anticipated.

Design work on the uni-axial panel-tensioning device was temporarily suspended in November for the more urgent items mentioned above. This work has now resumed, as indicated in Figure 2.2. The device is expected to be constructed in March and to be operational in April 1978.

CHAPTER 3

SPECIAL TEST SECTIONS

This chapter discusses the test sections used for testing panels at non-normal sound incidence. The design, construction, and initial testing will be described.

3.1 Objectives

In practice, aircraft fuselage panels are subject to noise from many different directions. Propeller noise from a wing-mounted engine, for example, strikes panels on the forward cabin at about 20° from normal, and rear cabin panels at up to 60° from normal. Circumstances of non-normal sound incidence will also occur in single engine airplanes. With these considerations in mind, it appeared desirable to obtain experimental sound transmission data for several angles of sound incidence.

The basic acoustic test facility at the KU Flight Research Lab is limited to testing panels at normal (perpendicular) sound incidence. The basic facility is described in detail in References 1 and 2. The facility was constructed with the intent of eventually testing panels at non-normal incidence. Thus the rail system was made long enough to accommodate special test sections which would hold flat panels slanted to incident sound. See Figure 3.1.

Four special test sections were then designed. Each section had to be able to support a flat panel at a different angle to incident sound. Each section had to be mechanically compatible with the existing basic facility, with regard to end connections and overall length. Also, the clamping conditions at the panel edges needed to be similar to the conditions obtained with the basic normal-incidence facility.

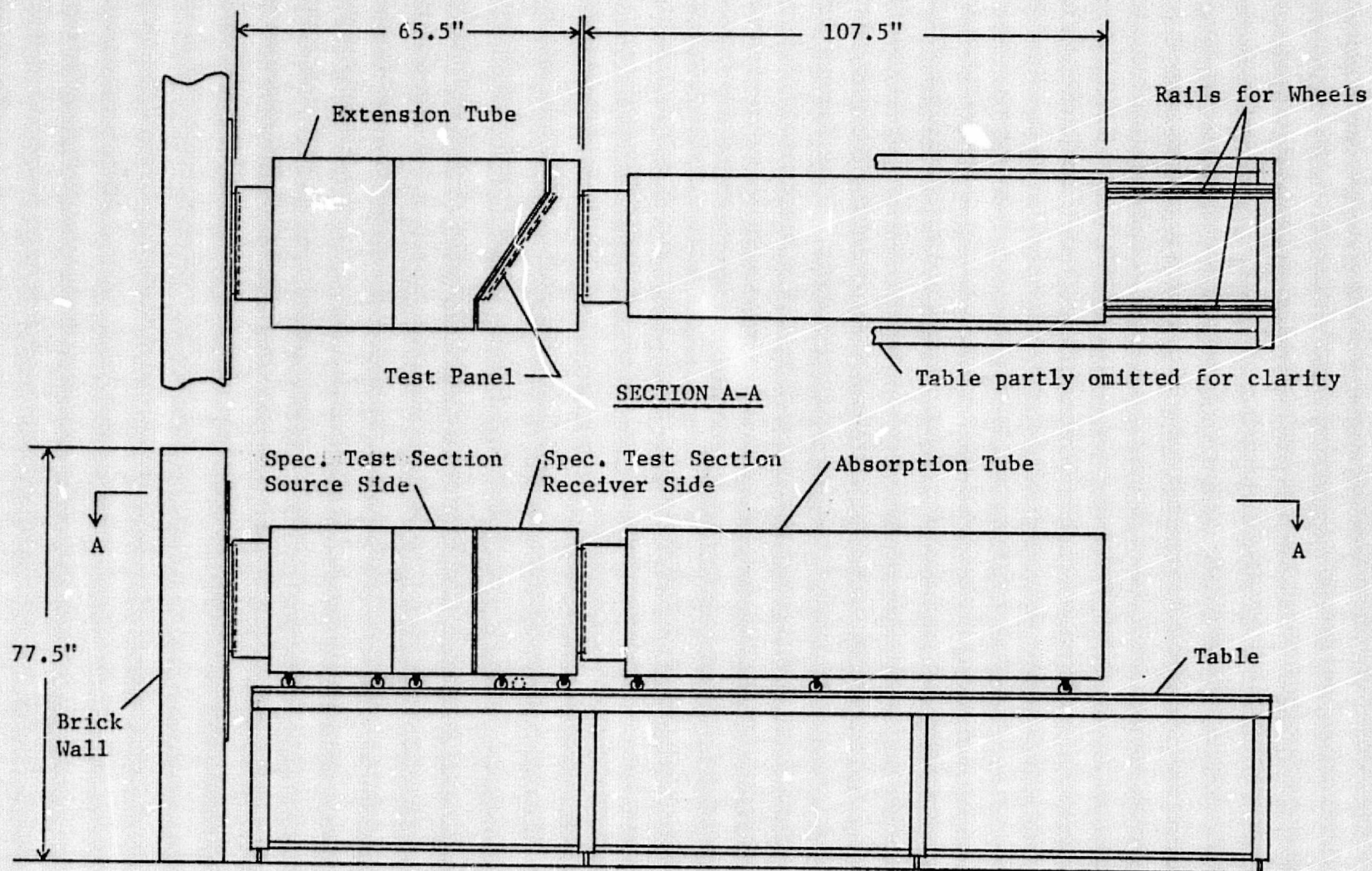


Figure 3.1: KU-FRL Acoustic Test Facility Showing Placement of Special Test Sections

Those clamping conditions are described in Section 4.2.

It was intended to test panels at angles of incidence of 15°, 30°, 45° and 60° from normal. The analysis of reflection patterns, shown in Figure 3.2, revealed that an undesired standing wave pattern would probably occur in the 45° test section. The undesired mode had a speaker-panel-sidewall-panel-speaker path. To avoid this eventuality, a 40° section was built instead.

To minimize acoustic reflections inside the test sections as far as possible, sound absorbing materials, including fiberglass wedges, line the inside of all sections, leaving an 18" x 18" square, axial sound path unobstructed. At this writing, about one half of all necessary absorbing materials have been installed.

3.2 Design Description

All the special test sections are made of pressboard, a hard, dense wood product. Various thicknesses were laminated before being glued and stapled together. All edge and corner joints were rabbeted, and numerous cleats were used, for high strength. Strength was considered important because clamping forces up to 800 lb per clamp were anticipated. All walls are one inch thick, and all areas of high stress are reinforced. The sections were built by Crestmark Woods, the same local (Lawrence) firm that earlier built the absorption tube used in the facility.

Figure 3.1 shows the positioning of the special test sections during testing. Note that the sections fit between the brick wall-encased sound source, and the absorption tube.

It was desired to have all test sections the same length so that any standing waves which arose would be similar for all sections.

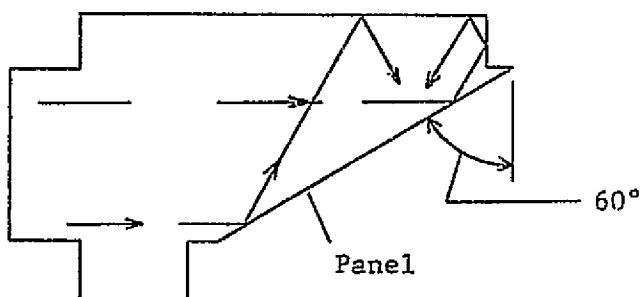
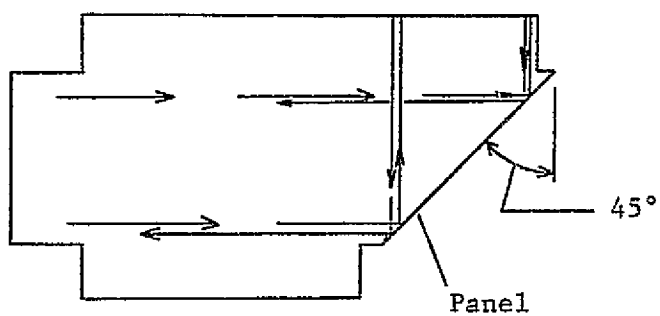
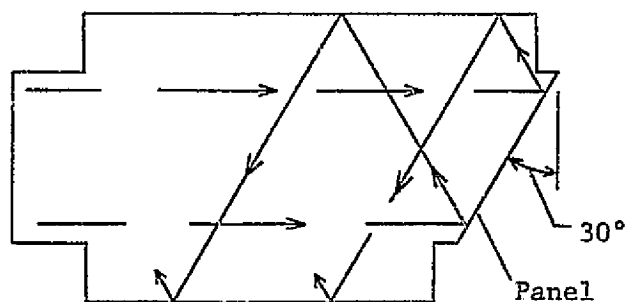
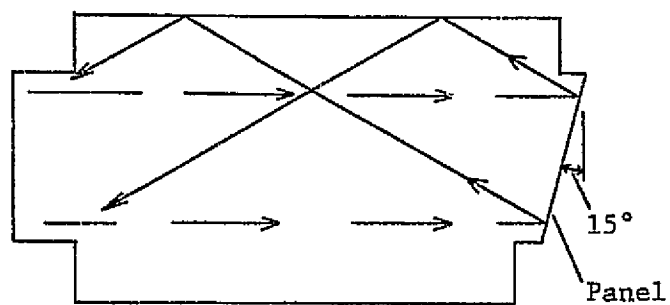


Figure 3.2: Sound Reflection Patterns in Four Prospective Non-normal Incidence Test Sections (Source Side)

This was recognized as an ideal, rather than fully achievable, goal because of the different source-to-panel lengths which result from different slant angles. Each complete test section assembly, except the 60° one, totals 65 inches in length. See Figure 3.3. Note that the extension tube is used with the 15°, 30°, and 40° test sections, but not with the long 60° test section.

The straight, 4-foot-long (average) sound path to the panel also ensures that very nearly plane waves impinge on the test panel. To take advantage of this planarity, the clamping system enables retesting of normal-incidence panels between a special test section and the absorption tube. It is planned to compare the results of such tests with previous tests made with the basic facility.

The clamping arrangements are shown in Figure 3.4. The eight U-bolts on the receiver side of the test section imitate the attachment loops on the speaker baffle plate. Thus the absorption tube connects to the special test sections exactly as it does, during the usual normal-incidence testing, to the brick wall. Similarly, adjustable hooks on the extension tube imitate those on the absorption tube. Thus, the extension tube connects to the brick wall exactly as does the absorption tube. The connection between the extension tube and the source sides of the special sections is simpler. Here, only four U-bolts and four over-center clamps are used because compatibility with the existing adjustable panel clamping system was not required.

3.3 Costs

In the proposed project budget, the special test sections were not listed specifically. The test sections were therefore paid for out of "Electronics" cost underruns, and the "Other expenditures" category.

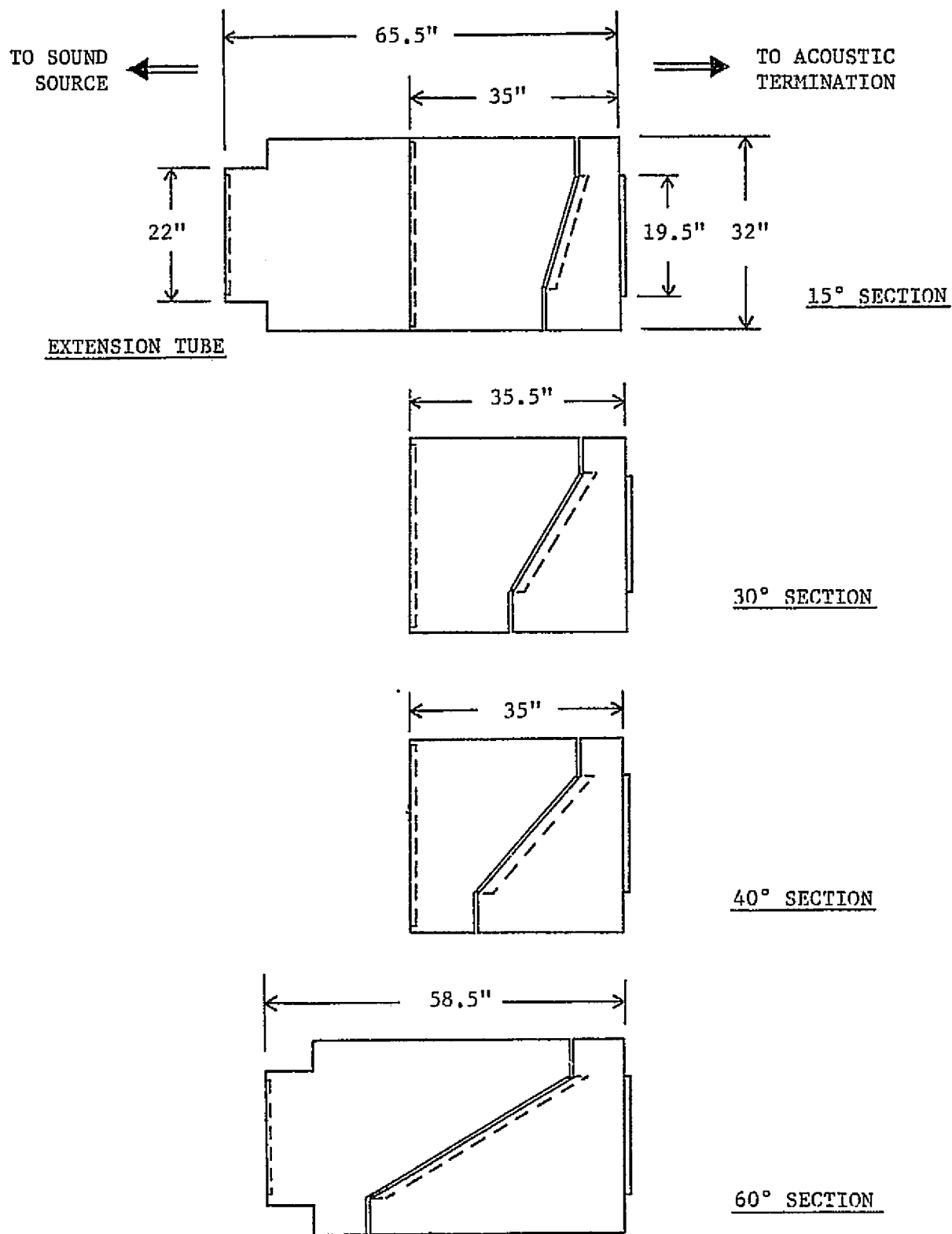


Figure 3.3: Top Views of the Special Test Sections
(Connecting Hardware not Shown)

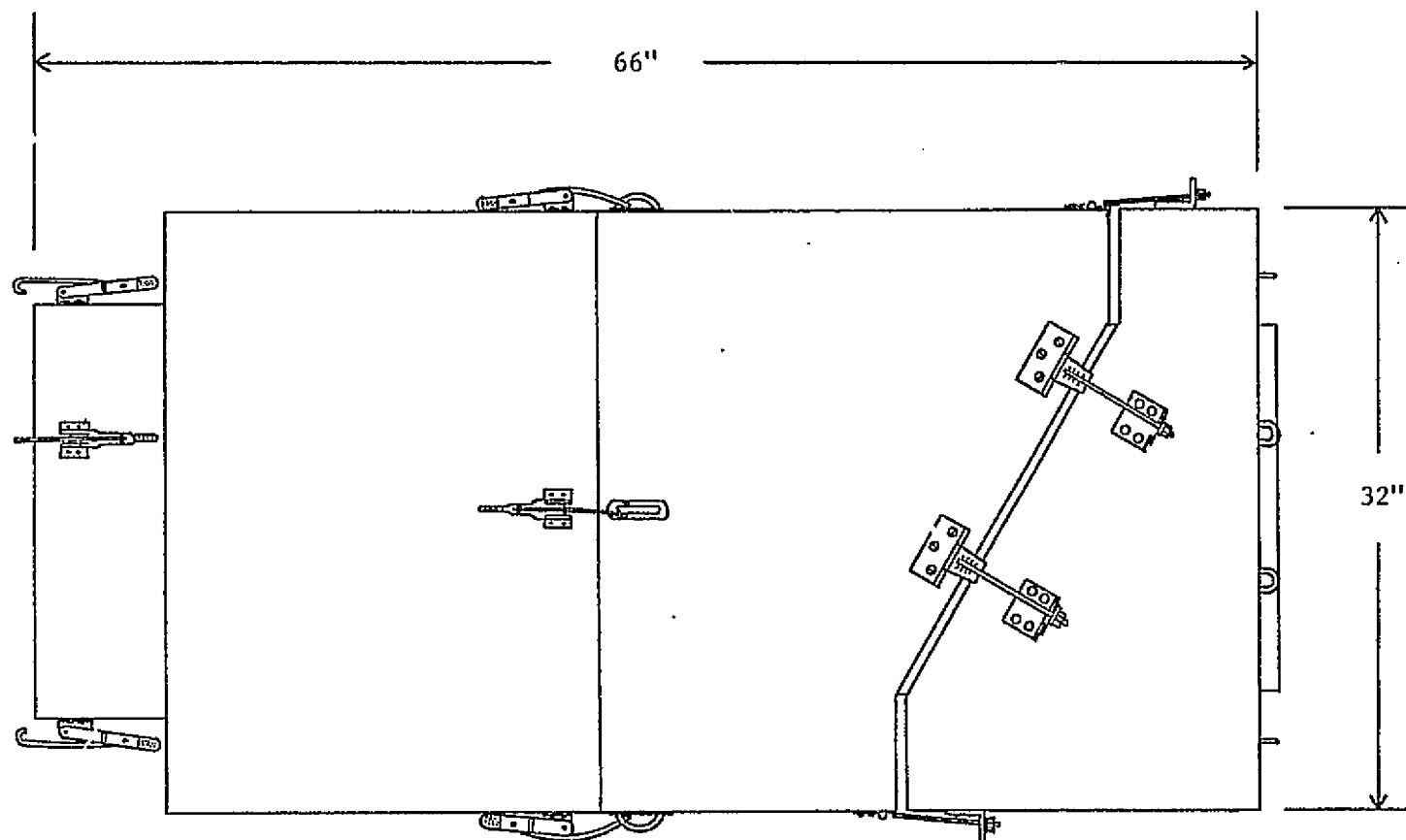


Figure 3.4: Clamping System of the Special Test Sections (Top View)

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Figure 3.5: 30° Special Test Section on Table
(Connecting Hardware not yet installed)

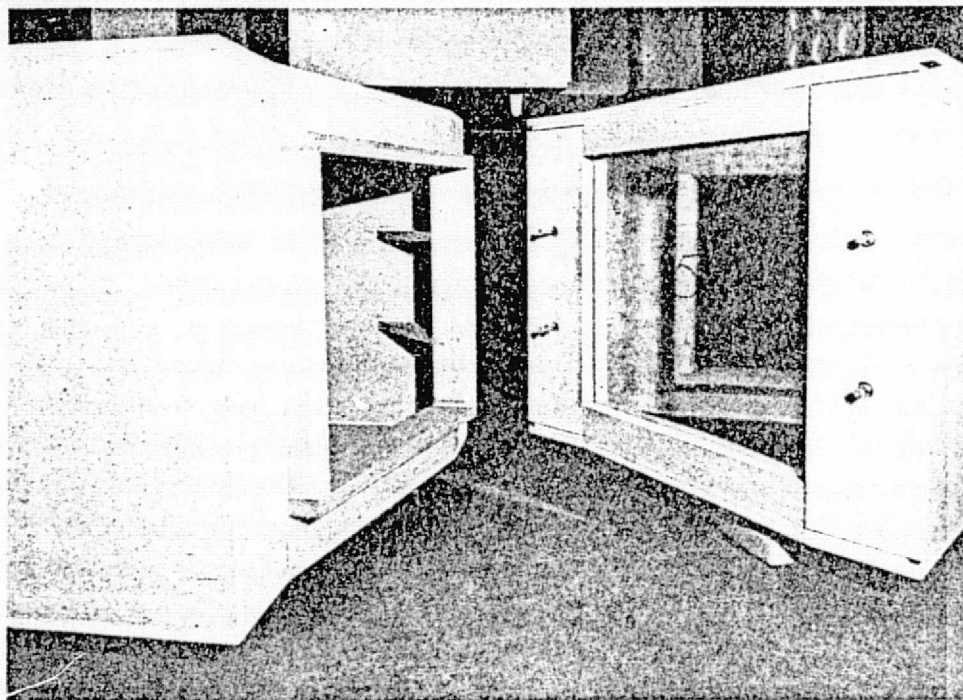


Figure 3.6: 15° Test Section in Normal Testing Configuration
(Source Side is on the left ; clamping hardware
is not yet installed)

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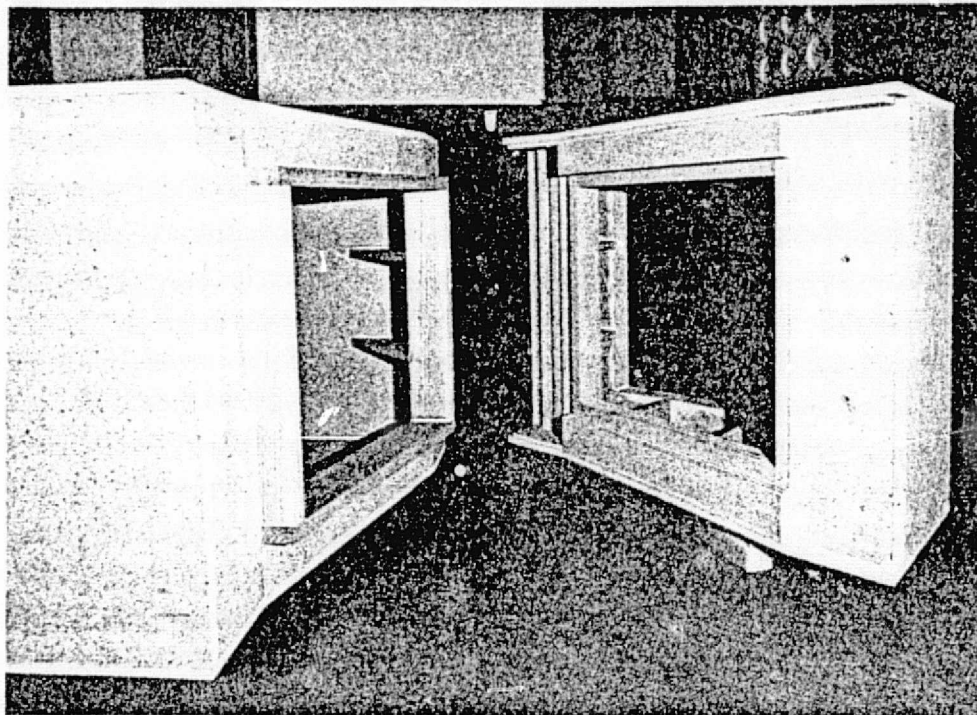


Figure 3.7: 15° Test Section in Uni-axial
Tension Testing Configuration
(Clamping Hardware not Installed)

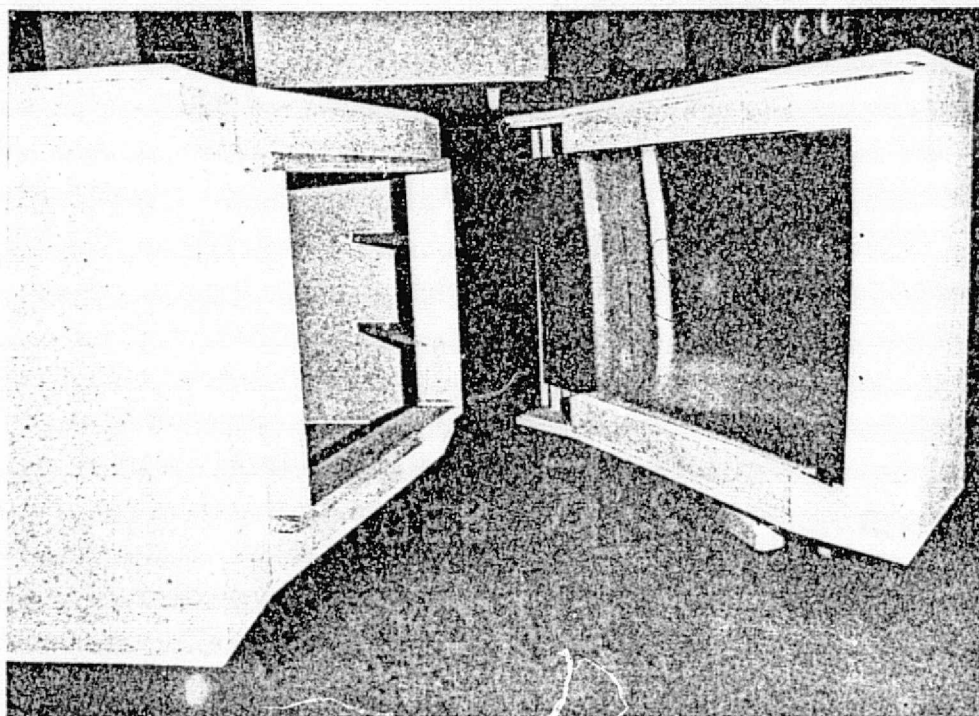


Figure 3.8: 15° Test Section in Uni-axial
Tension Testing Configuration
with Panel (Clamping Hardware not Installed)

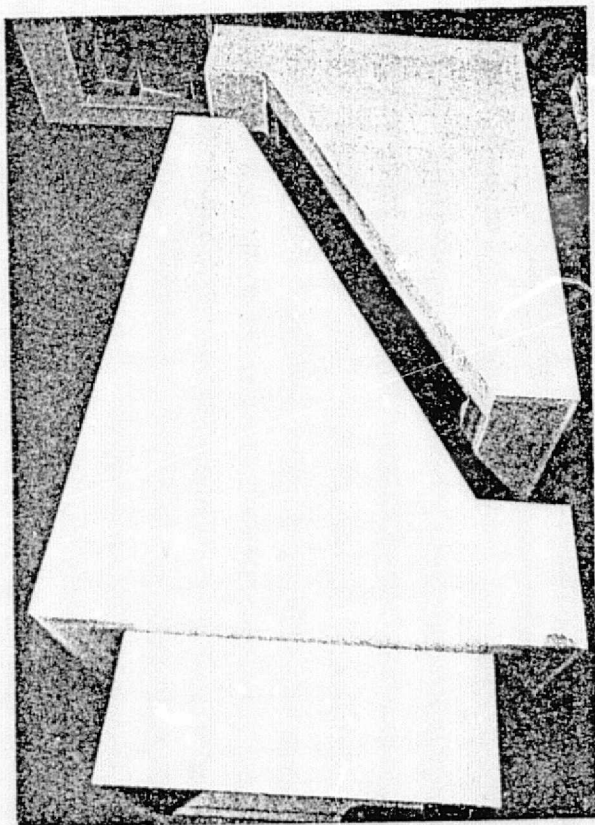


Figure 3.9: Top View of 60° Test Section (Clamping Hardware not Installed)

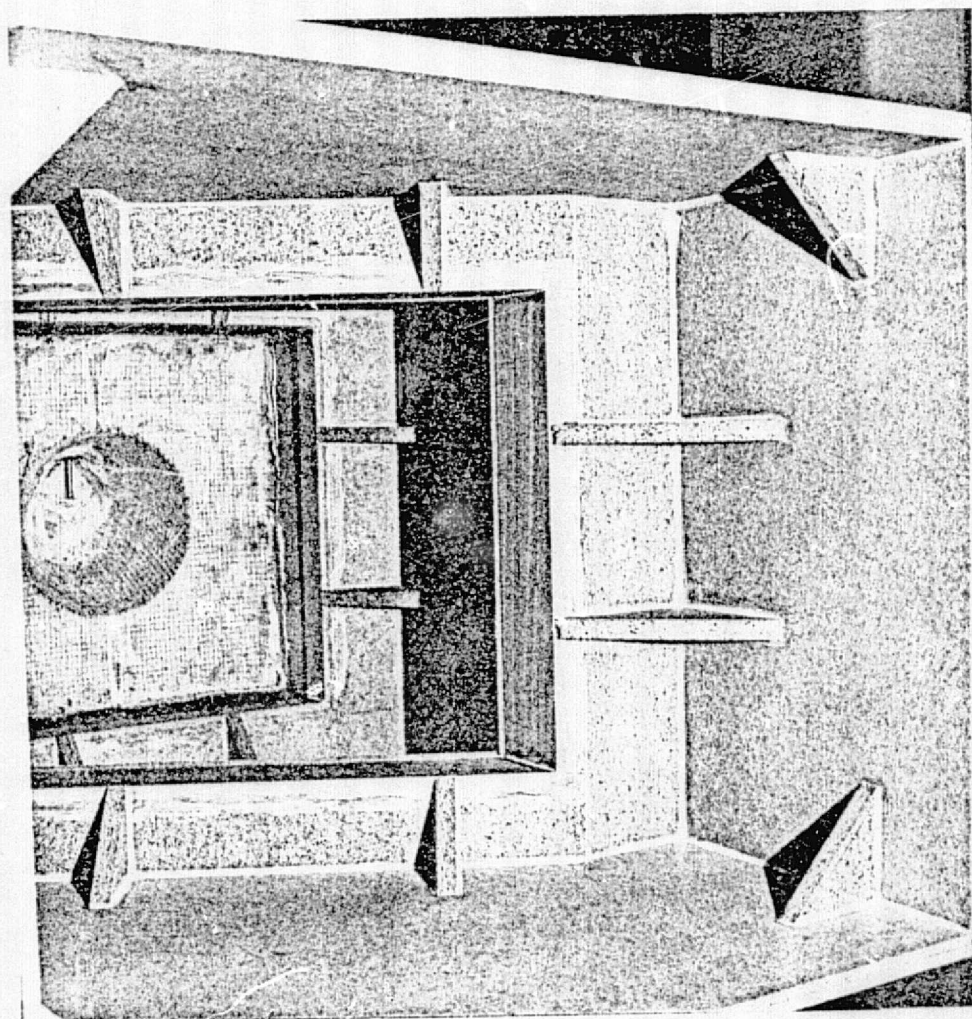


Figure 3.10: View Through Source and Receiver Sides of 30° Test Section into the Absorption Tube Showing Central Microphone Position. (Note: Absorbent fiberglass in Test Section and test panel are not shown.)

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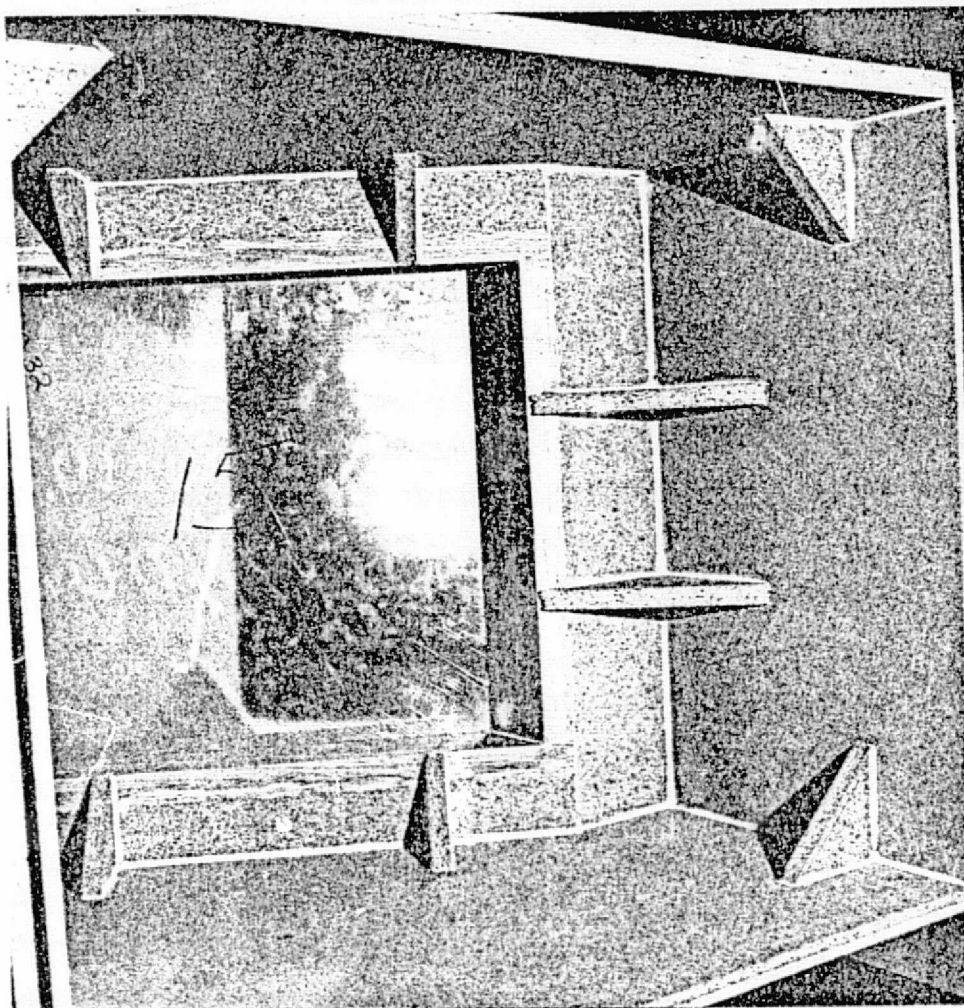


Figure 3.11: View into Source Side
of 30° Test Section with
Test Panel in Place.
(Absorbent Fiberglass not
yet installed in Test Section.)

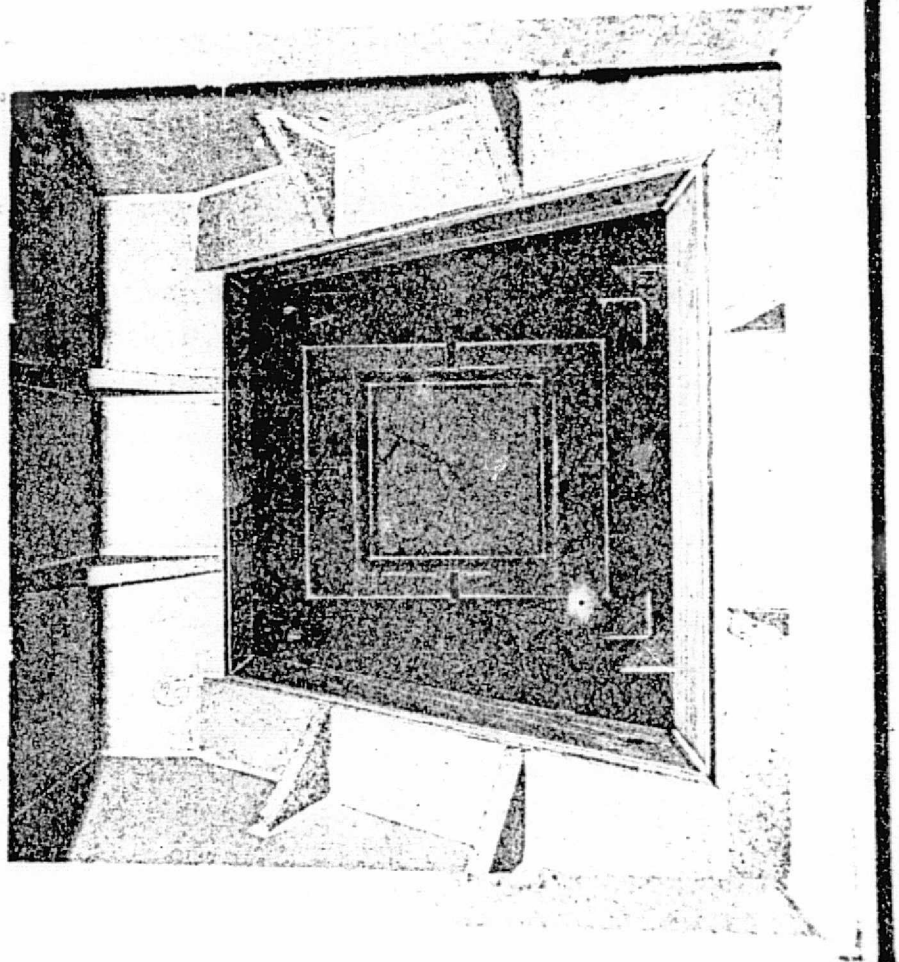


Figure 3.12: View Through Receiver Side and Source Side of 30° Test Section, and Extension Tube, Showing Foam Covered Loudspeaker Array and Source Microphone (Absorbent fiberglass not yet installed in Test Section, and Test Panel not in place)

The Crestmark Woods carpentry shop, whose quality work on the absorption tube was very satisfactory, submitted the following bid:

Extension tube	\$200
15° test section (2 pcs.)	700
30° test section (2 pcs.)	700
40° test section (2 pcs.)	700
60° test section (2 pcs.)	<u>800</u>
	\$3100

This bid was accepted. Accruing costs were monitored by a Noise Project member during construction. The total bill was exactly as bid.

Other non-salary expenses connected with the special test sections are shown below. As final preparation of the sections is not complete at this writing, cost estimations were necessary in certain categories as indicated.

Clamping hardware	\$134.84 (Est.)
Wheels	22.50
Sound absorbing materials	36.81 (Est.)
Glue	15.12
Transportation, incl. test sections	4.60
Silicone sealant	11.94
Stain, varnish, paintbrush	<u>21.12</u>
	Subtotal \$246.93
Bare test sections	<u>3100.00</u>
Non-salary total	3346.93

3.4 Calibration

Calibration will consist primarily of determining the best locations for the two microphones. Each microphone will be suspended about 15-30 cm (6-12 inches) from the panel. Initially, the microphones will be placed near the center of the test panel, and held in position by guy wires. Tests will be conducted with simple, bare metal panels, and the microphone locations will be adjusted until test results agree reasonably well with the mass law. From considerations of standing wave patterns, it appears likely that the best position will be about 5-15 cm radially from the center of the test section, in the direction of a side-top junction.

Other aspects of calibration will involve comparison of observed and calculated resonance frequencies. This may include impedance considerations with respect to both the slanted (non-square!) panels and the now-altered absorption cavity configuration.

CHAPTER 4

CHARACTERISTICS OF THE KU-FRL PANEL SOUND TRANSMISSION FACILITY

This chapter discusses the test conditions of the KU-FRL facility. It is important to know and understand the conditions of acoustic tests because those conditions can strongly influence the test data. The four types of conditions addressed in this chapter appear to be the most important ones in this facility.

4.1 Microphone Position

Test data will be influenced by microphone location if the sound field is not reasonably uniform. In the present closed (cavity) system, non-uniformities can be expected when standing waves are present. See Reference 4.

On the source (loudspeaker) side of the panel, about nine inches of flexible plastic foam surround the "source" microphone on four sides. This attenuates the sound waves before and after reflection from the cavity sides. As a result, standing waves in these two directions are severely weakened and, in fact, are not observed at all. In the third direction--axially--the wall separation is only one inch so that the lowest frequency standing waves will occur at about 13,000 Hz. This is well above the frequency range of present interest. When tests are performed with the special slanted panel test sections, however, the axial distance will be much larger. The potential standing waves in this case may produce temporary difficulties until a suitable microphone position is found.

On the receiver side, the measuring microphone is backed by eight feet of various absorbing materials to minimize axial reflections

and concomitant standing waves. Above, below, and to the sides, however, the absorbing material is only about 5 inches thick, and is loosely packed. Standing waves in these directions may be rather strong.

The frequency of standing waves between two parallel reflecting surfaces is:

$$f = \frac{c_0}{2} \frac{m}{a} \quad (4.1)$$

where m is the mode number. If another set of parallel surfaces, perpendicular to the first set, is present, higher order standing waves are possible. These would take the form of a diamond shaped reflection pattern in which all reflections occur at angles of 45° . In the present facility standing waves of this type will probably be greatly weakened by passing through four feet of absorbing material during each circuit. Therefore, simple one-dimensional standing waves may be the most noticeable.

Figure 4.1.1 shows the sound pressure patterns associated with several standing wave modes. The difficulty in determining the predominant mode in the KU-FRL absorption tube is that observations of sound pressure patterns indicate one mode, and observations of the affected frequencies indicate another. Limited observations of low sound pressure on the axial centerline suggest the presence of the 1,1,0 mode (see Figure 4.1.1). The frequency apparently most strongly affected (note spikes in IL^* curves, Figures 4.3.4, 4.3.5, 4.3.6), however, is between 700 and 750 Hz. This frequency suggests occurrences of the 2,0,0 mode between parallel walls spaced 45.72 cm (18 in.) apart. This sound pressure-frequency anomaly is presently under study.

* IL = Insertion Loss (see Section 4.3.3).

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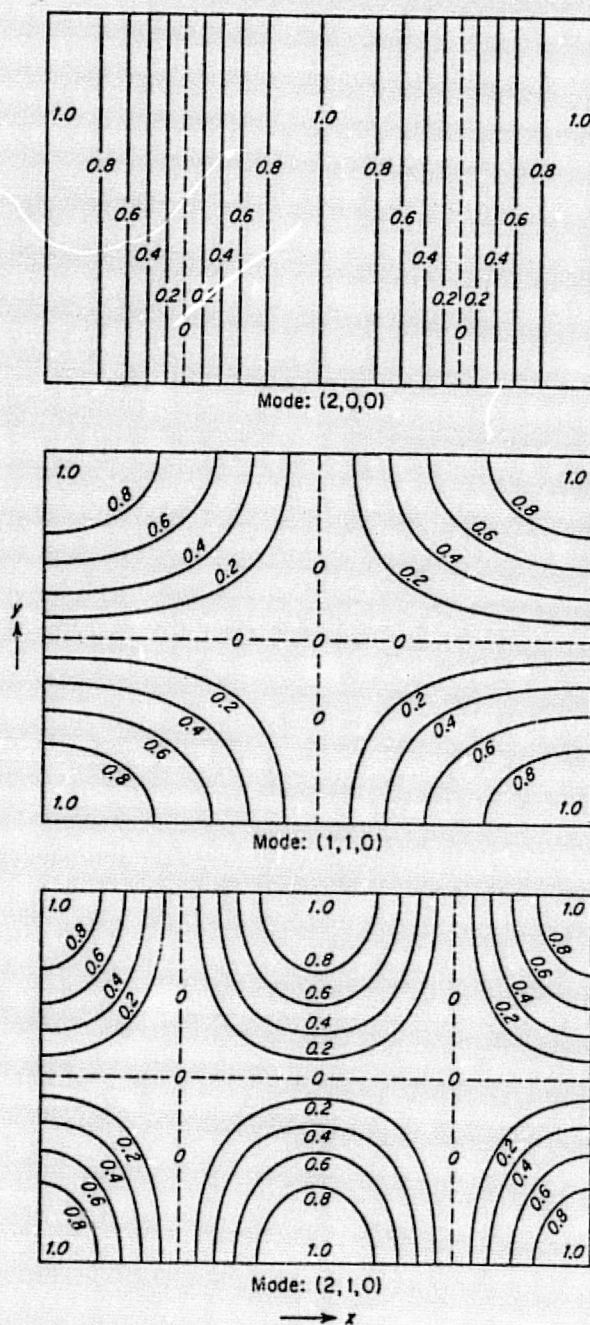


Figure 4.1.1: Contour Plots of Relative Sound Pressure in a Section Plane Through a Rectangular Cavity, for Three Different Modes of Vibration. (From Ref. 4.)

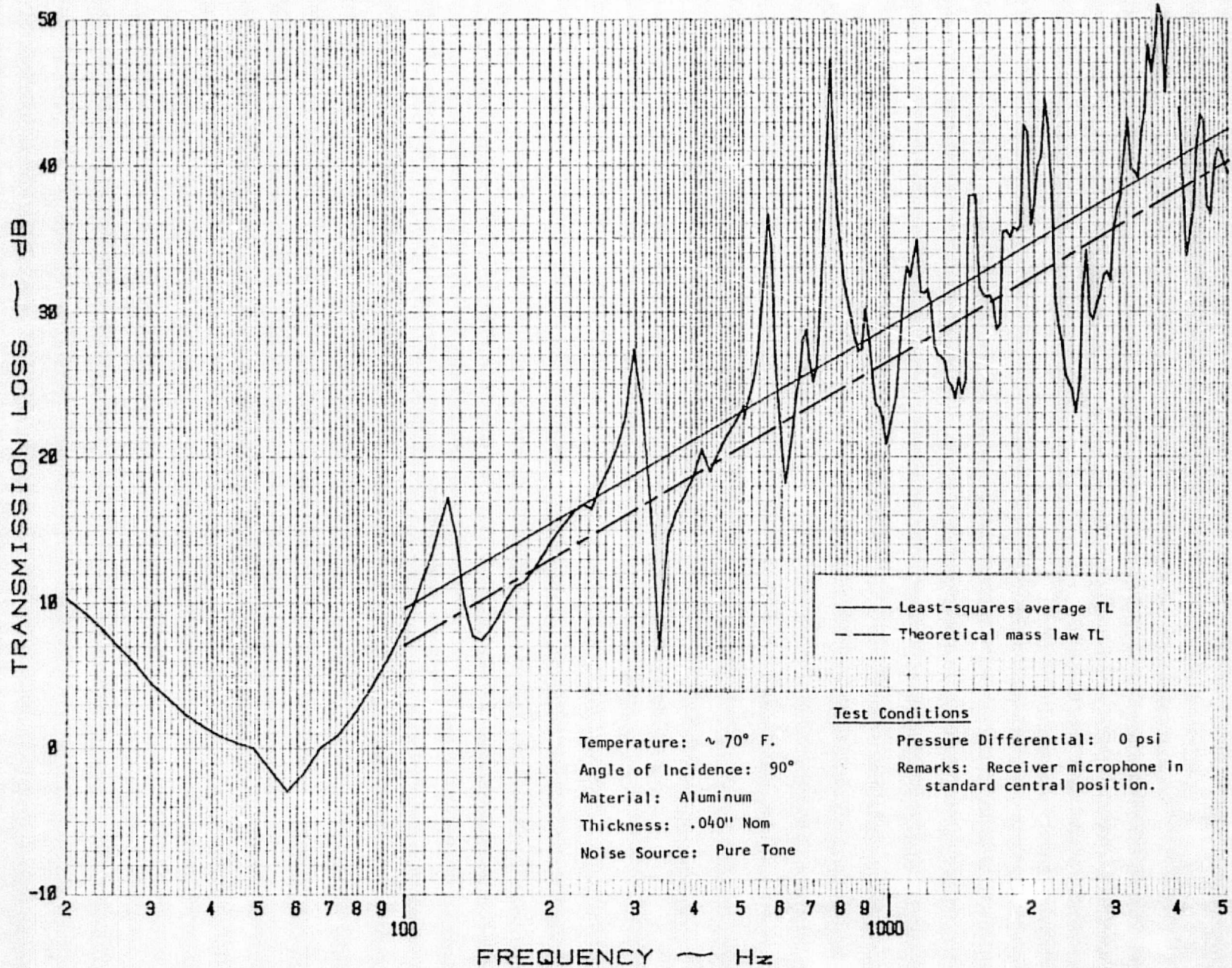
Experimental TL curves for three receiver microphone positions are shown in Figures 4.1.2 - 4.1.4. Figure 4.1.2 gives the results with the microphone in the standard central position, 20 cm (8 in.) from the test panel. This is the position used for the data of Appendix B, and Reference 3. Figure 4.1.3 gives results obtained with the microphone still on the axial center line, but now only 6 cm (2 3/8 in.) from the panel. For Figure 4.1.4, the microphone was 6 cm from the panel, and 12 cm (4 3/4 in.) diagonally (45°) off the axial center line. All tests were performed with a bare .040 in. Aluminum panel.

In the standard position the least-squares average observed TL closely matches the theoretical 6 dB/octave slope in the mass law region. The observations are about 2.7 dB too high, however.

The second position, a central location close to the panel, shows an experimental slope of only 4 dB/octave. Note, however, that the possible standing-wave peak at 700 - 750 Hz has been weakened, as have other possible standing-wave peaks at about 1800 and 3600 Hz.

In the third position, away from the possible standing-wave node location, both slope (5.4 dB/octave) and overall TL magnitude agree reasonably well with the mass law. This position, or one similar to it, may be preferred if more accurate absolute panel data is desired. Such an off-center position may be close to that employed in the tests of Reference 5, as suggested by the text therein.

It should be kept in mind that the standard central position must be retained to compare subsequent panel test data with the data already obtained from the present facility.



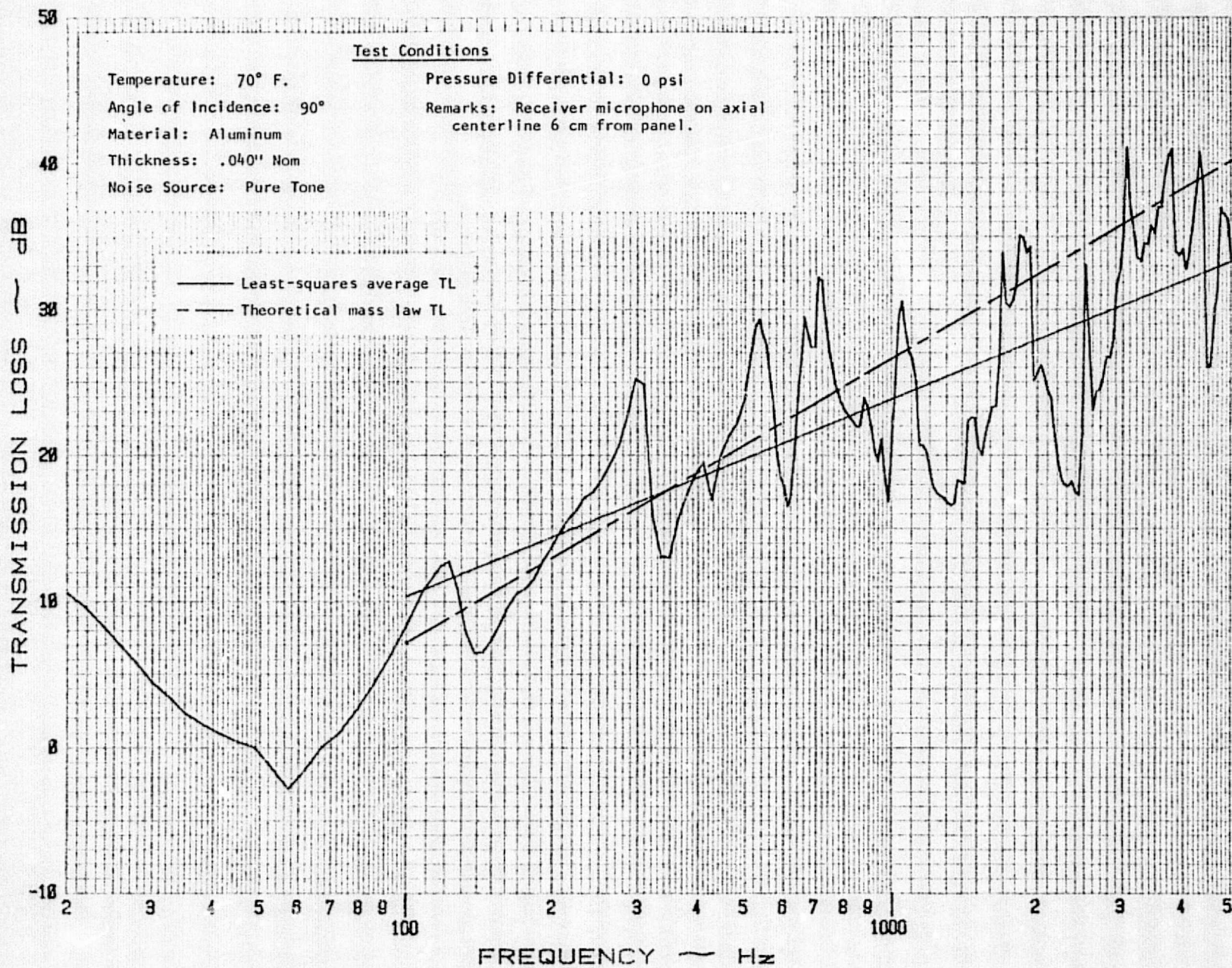
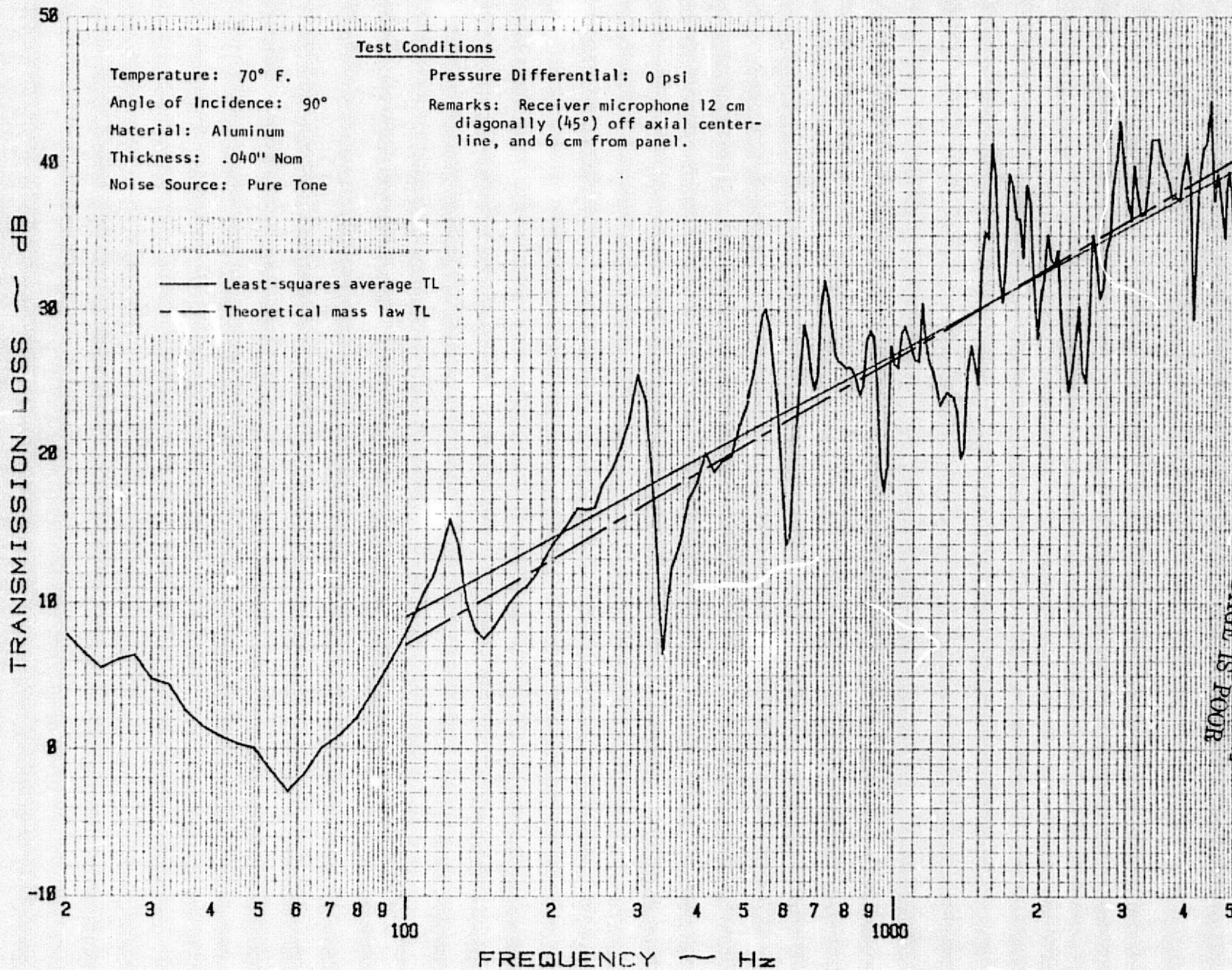


Figure 4.1.3: Experimental Sound
Transmission Loss, Receiver Micro-
phone Position Changed

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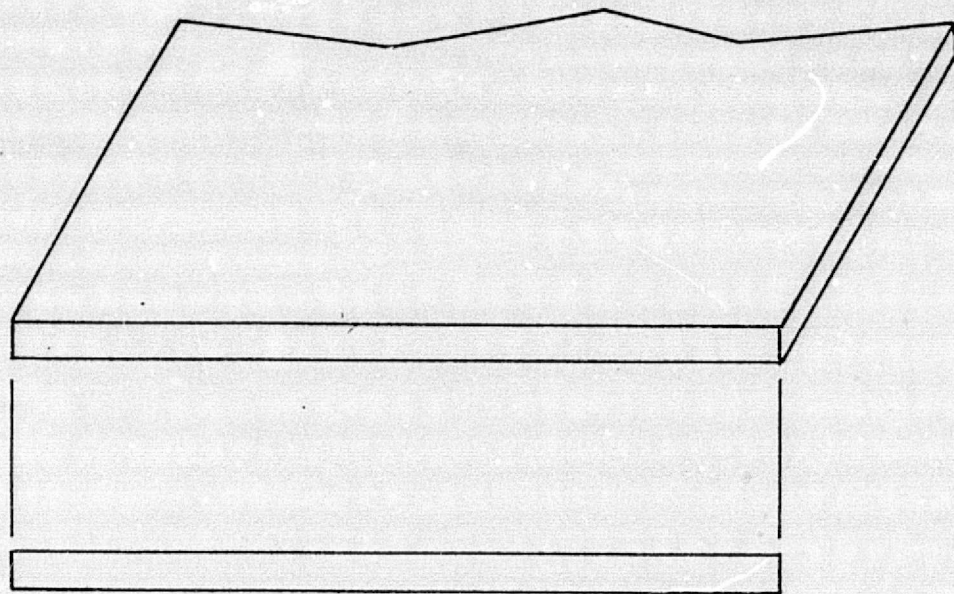
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4.2 Panel Edge Conditions

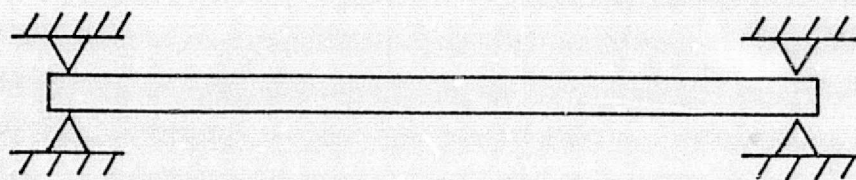
There are three basic types of plate edge conditions, free, simply supported, and clamped. These are represented in Figure 4.2.1. Intermediate conditions are also possible, and are often encountered in practical situations. Examples of intermediate conditions are represented in Figure 4.2.2.

The present investigation is intended to deal only with rectangular panels having all edges clamped. Therefore, the test facility was designed to clamp all edges of test panels. Reasonable care was taken to ensure flat, parallel clamping surfaces. During checkout of the pressurization system (see Reference 1), a 1/8" thick gasket of fabric-reinforced rubber was attached to the speaker-side clamping surface. This was found necessary to satisfactorily seal the pressurization (actually vacuum) system. In the interest of testing all panels under identical edge conditions, this gasket has been left in place during all regular tests. On the absorption tube, the clamping surface is varnished end-grain wood.

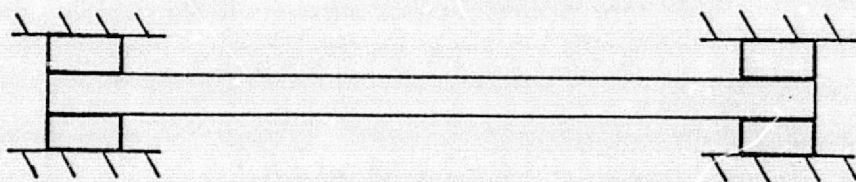
Initial investigations showed that resonance frequencies (as observed in TL curves) varied considerably with the clamping force applied. As time was not available to track down the ultimate cause of this effect, several tests were performed to establish a reasonable level of force which would be used for all panels, to ensure the uniformity of test conditions. The selected level was 3100 lbs total force, corresponding to 25 in-lb of nut-tightening torque on each of the eight clamps. The torque-axial force relationship of Reference 6 was used. Figure 4.2.3 shows the clamping surface over which this force is distributed.



Free Edges

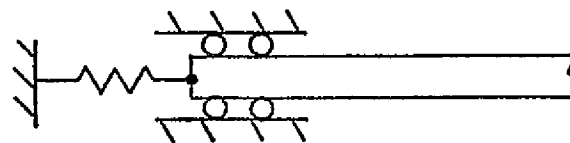


Simply Supported Edges

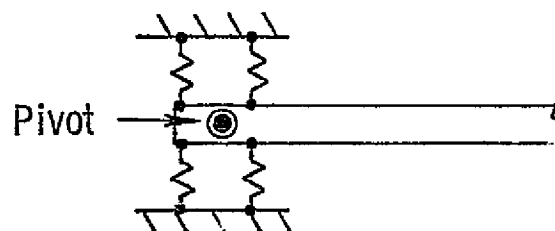


Clamped Edges

Figure 4.2.1: Basic Types of Plate Edge Conditions



Translation Permitted against a Spring Force,
but No Rotation



Rotation Permitted against a Spring Moment,
but no Translation

Figure 4.2.2: Intermediate Types of Plate Edge Conditions

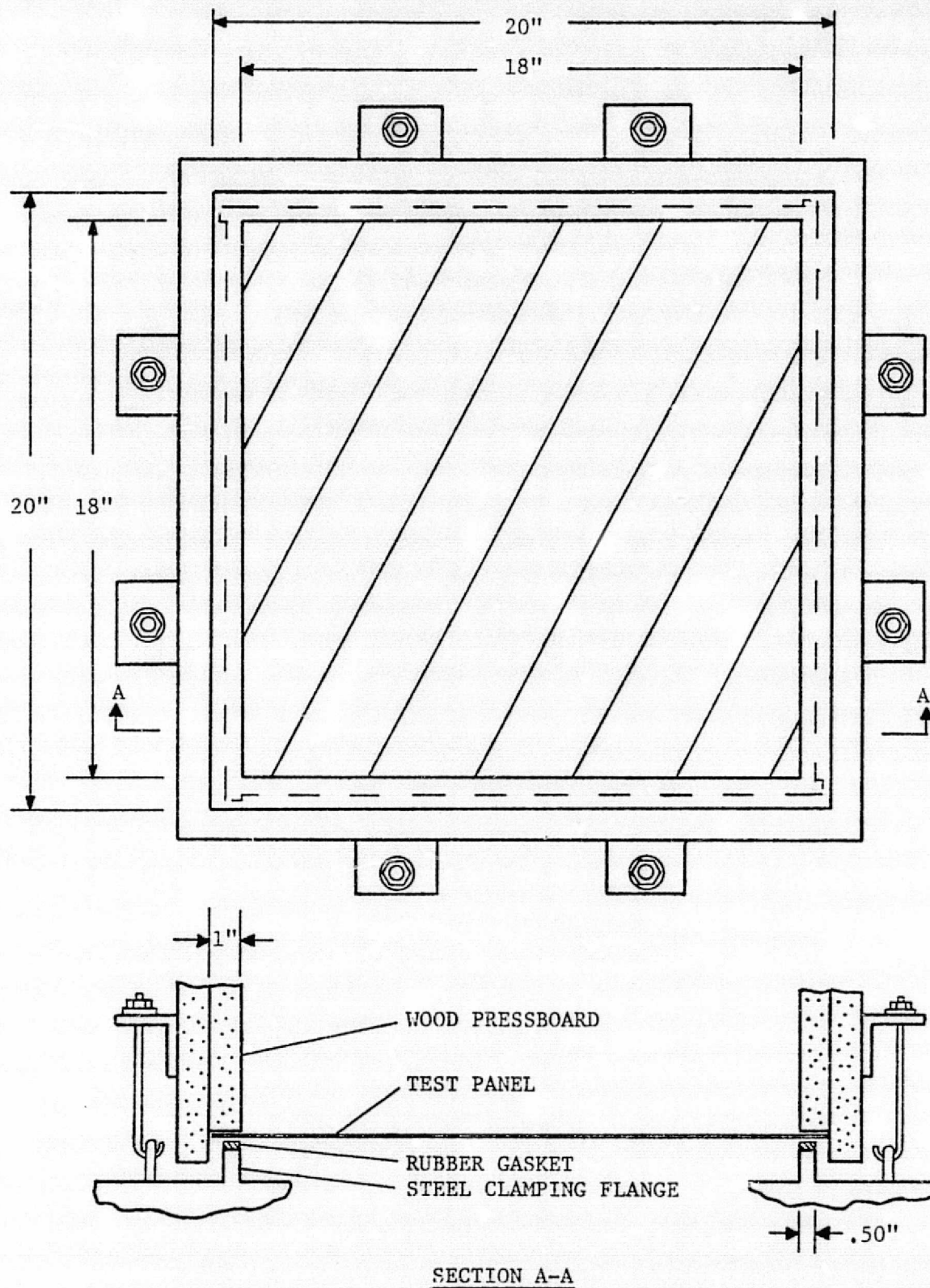


Figure 4.2.3: Panel Clamping Arrangement

As part of the investigation of the edge conditions actually obtaining (in the tests), the rubber gasket was replaced with four 1/4-inch thick aluminum bars. Data obtained with this relatively inelastic and damping-free clamping surface were very similar to rubber gasket data. This tended to eliminate mechanical damping, due to the rubber gasket, as the main cause of the resonance frequency variation. During these tests it was noticed that one aluminum bar tended to be especially loose and occasionally fell from its glued position, when the test section was opened after being uniformly clamped with 3100 lb force. When the bar did not fall, the surface of the adjoining bar was found to be up to .040 in. from flush with it. It was concluded that the clamping surfaces were not sufficiently flat and parallel to produce uniformly clamped edges. This conclusion is supported by the louder-than-previous rattling sounds which were audible each time the sweeping frequency excitation coincided with a resonance frequency. Previous testing had been performed with the rubber gasket in place.

A part of one "clamping" edge is thought to be slightly low. In tests with the aluminum-bar edges, the 1,3 and 3,1 modes coincided fairly well only at the highest force tested, 4300 lbs. When the rubber gasket is used, the 1,3 and the 3,1 modal frequencies (as observed on TL plots) coincide at much lower clamping forces. This suggests that the rubber gasket conforms to the slightly uneven clamping surface, thereby supporting the panel all along the edge, at the "standard" 3100 lb clamping force.

The edge condition produced by the rubber gasket and 3100 lb force cannot be easily characterized as either clamped or simply

supported. It appears to be an intermediate condition. In tests using up to 12,000 lb clamping force, the 1,1 mode resonance frequency showed no tendency to asymptotically approach a limiting frequency, although the 1,1 resonance frequency did increase steadily as the clamping force was raised. This suggests that the rubber gasket is behaving as a torsional spring which permits changes in slope at the edge of the panel. Very high clamping forces are apparently required to produce a spring force large enough to approximate a truly clamped edge condition. Of course, a high effective shear stiffness is also required to prevent the panel edges from being pulled in toward the center as the panel vibrates, especially in the 1,1 mode. This effect may also play a role in the intermediate edge condition present.

The edge condition, regardless of its exact nature, appears to influence panel vibration modes differently. In going from an ideal, simply supported panel to a perfectly clamped panel, the lower mode frequencies experience larger percentage changes than the frequencies of higher modes. Table 4.1 shows this, using data from Reference 7. Edge conditions which are not uniform all along the periphery of a panel also influence vibrational mode frequencies to different degrees. Table 4.2 presents data from Reference 8 for the partly simply supported, partly clamped panel of Figure 4.2.4.

The edge conditions of Figure 4.2.4 may be similar to those of the present investigation. Both panels can be considered clamped on all edges, but with regions near one or more corners where the panel is simply supported. Considering resonance frequencies, the 1,1 mode of such a panel behaves as if very nearly clamped, while higher modes approach simply supported behavior. For this reason, and because no

Table 4.1 Non-dimensional Natural Frequencies $\omega a^2 \sqrt{\frac{\rho h}{D}}$,
i.e. λ , of Square Plates

<u>Mode</u>	<u>All Edges Simply Supported</u>	<u>All Edges Clamped</u>
1,1	19.74	35.984
1,3	98.70	131.63
3,3	177.65	219.89
1,5	256.61	309.13
3,5	335.57	393.46
5,5	493.48	561.47
1,7	493.48	565.47
3,7	572.44	647.54
5,7	730.35	810.50
7,7	967.22	1061.53
1,9	809.31	900.84
3,9	888.26	981.24
5,9	1046.18	1146. *
7,9	1283.05	1313. *
9,9	1598.88	1719.5 *

* Estimate

Source: Reference 7.

Table 4.2 Non-dimensional Natural Frequencies of
Partly Clamped Square Plates

<u>Mode</u>	λ <u>(partly clamped)</u>	<u>Equivalent</u> <u>Percent Clamped</u>	<u>Equivalent Percent</u> <u>Simply Supported</u>
1,1	35.291	95.7%	4.3%
1,2	67.432	76.8%	23.2%
2,2	91.380	43.6%	56.4%

Source of Data: Ref. 8.

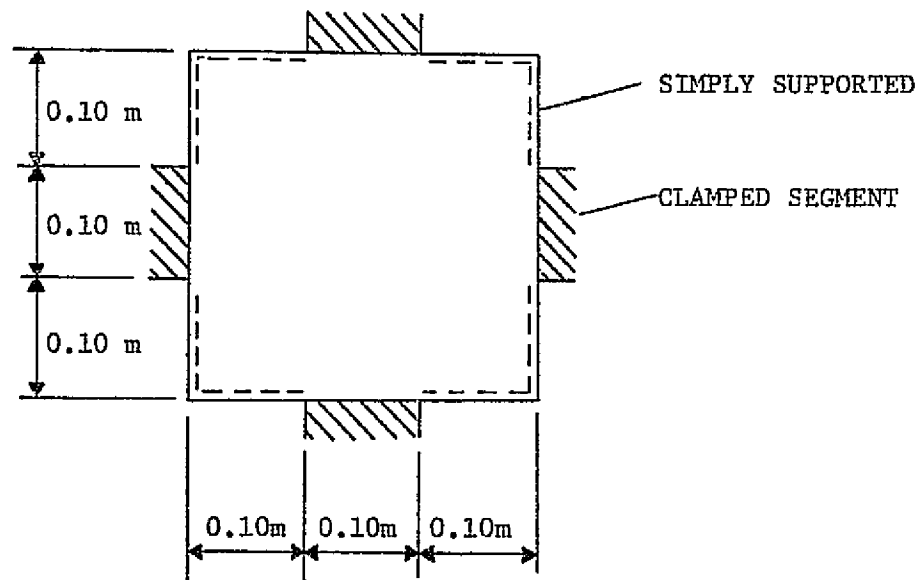


Figure 4.2.4 The Partly Clamped Panel of Table 4.2

additional data are available, test panels are assumed to be completely clamped in the 1,1 mode, and 50% clamped, 50% simply supported in higher modes, in Section 4.3. This assumption will be used to obtain approximate resonance frequencies for the test edge conditions, which are not known precisely. It will be seen that possible consequent errors can be neglected for the present purposes.

4.3 Cavity Influences on Sound Transmission Data

When panel testing began in the KU-FRL facility, several apparent anomalies were observed. The measured resonance frequencies of most test panels did not agree well with the calculated resonance frequencies. Negative transmission loss was also observed. This means that the receiver microphone registered a higher sound level, at some frequencies, than the source microphone. These problems are addressed in the present section.

4.3.1 Method

An investigation of the literature suggested that the KU-FRL test facility was of a type whose operation is imperfectly understood. References 9, 10, 11, and 12 discuss panel sound transmission into a hard-walled cavity; and References 13 and 14 discuss acoustic characteristics of cavities possessing an absorbent boundary. The KU-FRL facility will be analyzed by combining these two simpler cases.

Before continuing, acoustic quantities need to be discussed. Analogies with electrical circuits, though sometimes confusing, are so useful as to be nearly obligatory. Table 4.3 summarizes the acoustic equivalents of resistance, capacitance and related quantities.

To avoid errors in determining acoustic quantities, care must be taken that units agree, as a rigorous, consistent system of acoustic units has only recently been established.

If the KU-FRL test facility is analogized to an electrical circuit, then maximum power transmission will occur when the panel impedance matches the cavity impedance. Likewise, minimum power transmission will take place when the impedance mismatch is greatest. Another quantity, the panel/cavity coupling coefficient, also plays a role; but the degree to which impedances match appears to be the primary determinant of panel

Table 4.3 Acoustical System of Units
Used in this Report

<u>Quantity</u>	<u>Symbol</u>	<u>Units</u>	<u>Electrical Analogue</u>
Pressure	P	$\frac{N}{m^2}$	Voltage
Volume	V	m^3	-
Impedance	Z	$\frac{N \text{ sec}}{m^3}$ or mks rayl	Impedance
Acoustic Resistance	r_A	mks rayl	Resistance
Acoustic Reactance	X	mks rayl	Reactance
Acoustic Capacitance	C_A	$\frac{m^3}{N}$	Capacitance
Compliance	-	$\frac{m^5}{N}$	

Note: The system of Ref. 9 has been used as far as possible
for commonality.

transmission loss. Details of this theory of sound transmission will be considered next.

4.3.2 Impedances and Coefficients

The impedances of the acoustical "circuit" elements--the panel and the cavity--must be determined. From Reference 9, the impedance of a rectangular panel vibrating in the q,r mode is:

$$Z_{qr}(\text{panel}) = j \frac{\rho h}{\omega} (\omega^2 - \omega_{qr}^2) + \rho h \frac{\omega_{qr}^2}{\omega} \delta \quad (4.2)$$

The meanings of all symbols are given in the list of symbols at the beginning of this report. For simply supported plates, the natural frequencies are given by

$$\omega_{qr} = \pi^2 \left[\left(\frac{q}{b} \right)^2 + \left(\frac{r}{c} \right)^2 \right] \sqrt{\frac{D}{\rho h}} \quad (4.3)$$

For clamped panels, the calculation is more involved, and the use of tabulated values from Reference 7 is recommended. Figure 4.3.1 and Table 4.1 give the nondimensional natural frequencies of the modes most important in sound transmission. To dimensionalize these frequencies, the following relationship is used.

$$\omega_{qr} = \frac{\lambda_{qr}}{a^2} \sqrt{\frac{D}{\rho h}} \quad (4.4)$$

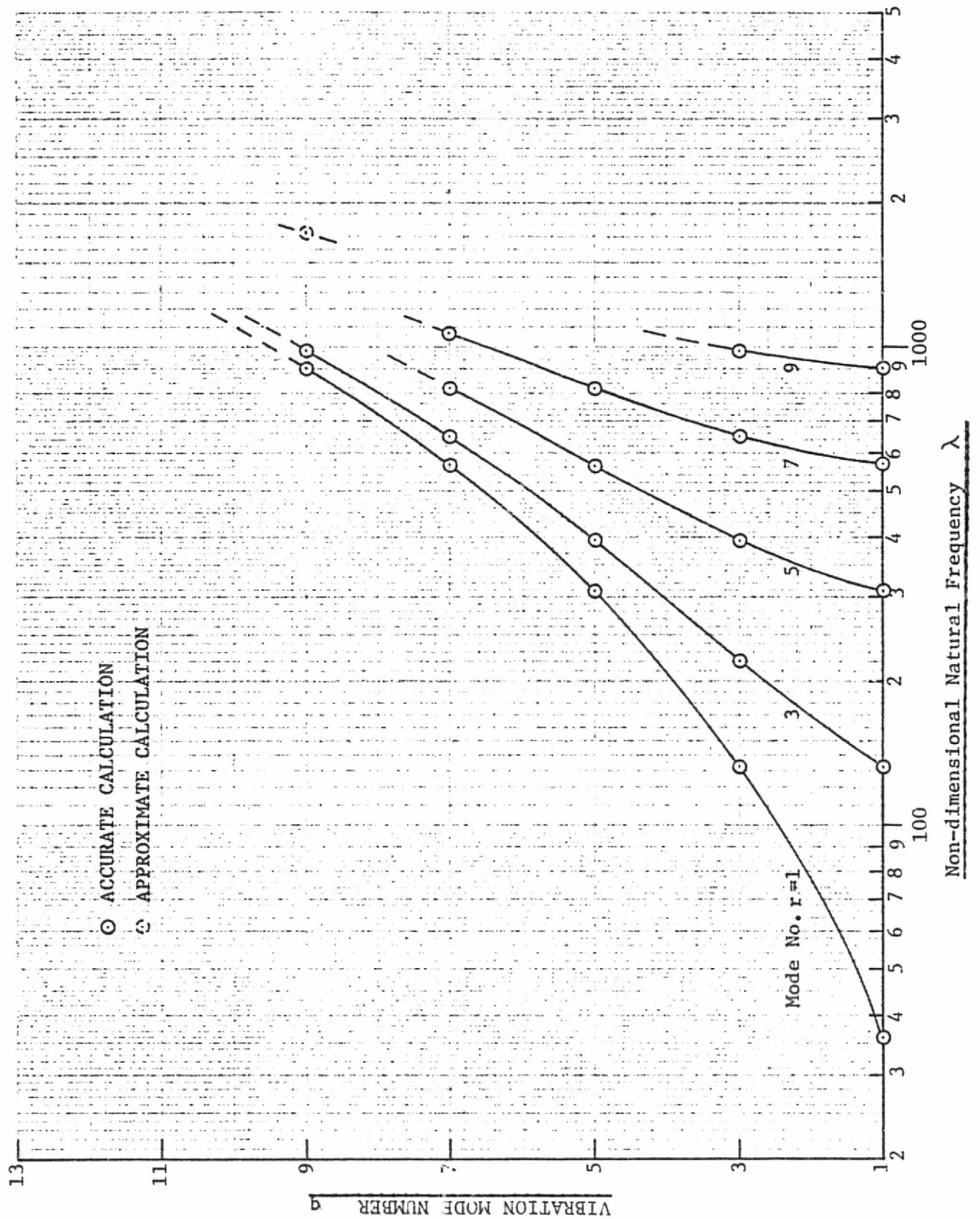
Reference 9 gives the cavity modal impedance as

$$Z_{mn}(o) \text{ cavity} = \frac{j \rho_o c_o \omega}{\sqrt{\omega_{mn}^2 - \omega^2}} \coth \left(\frac{a}{c_o} \sqrt{\omega_{mn}^2 - \omega^2} \right) \quad (4.5)$$

for the system of Figure 4.3.2. If $\omega_{mn} < \omega$, this expression becomes:

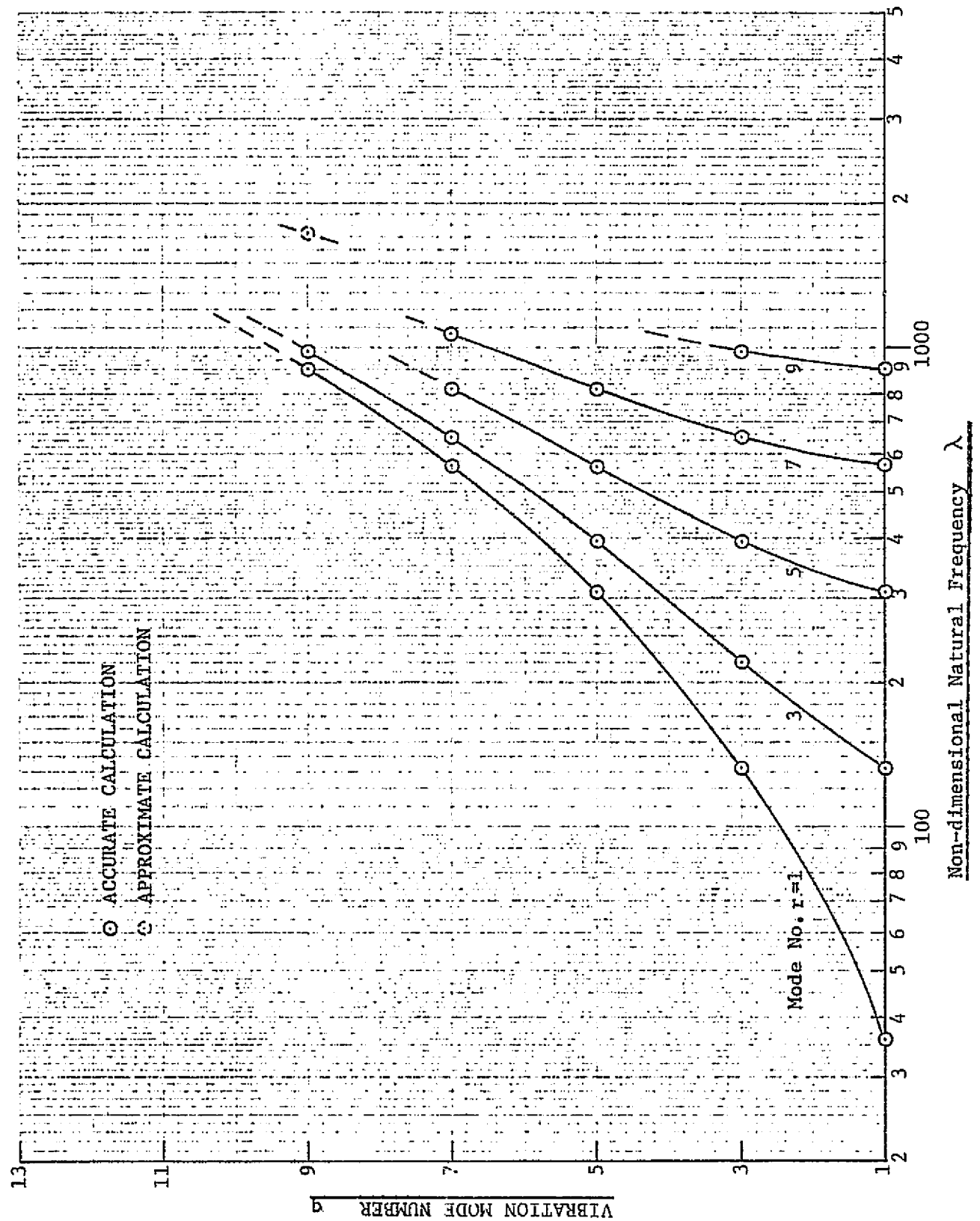
$$Z_{mn}(o) \text{ cavity} = \frac{-j \rho_o c_o \omega}{\sqrt{\omega^2 - \omega_{mn}^2}} \cot \left(\frac{a}{c_o} \sqrt{\omega^2 - \omega_{mn}^2} \right) \quad (4.6)$$

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CALC			REVISED	DATE	Figure 4.3.1: Non-dimensional Natural Frequencies of Clamped Square Plates	UNIVERSITY OF KANSAS	PAGE 42
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Figure 4.3.1: Non-dimensional
Natural Frequencies of Clamped
Square Plates

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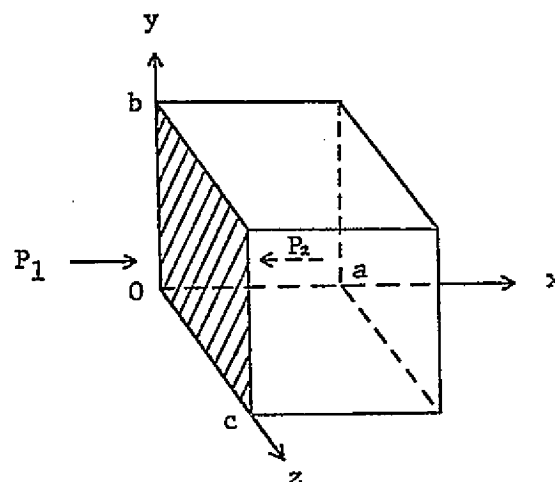


Figure 4.3.2: Cavity/Panel Geometry
From Ref. 9.

The cavity resonance frequency for the m,n mode is

$$\omega_{mn} = \pi c_o \sqrt{\left(\frac{m}{b}\right)^2 + \left(\frac{n}{c}\right)^2} \quad (4.7)$$

For a non-conservative (energy dissipating) system, the impedance is complex:

$$Z \text{ (cavity)} = R + jX \quad (4.8)$$

Here, X is the reactive impedance, as calculated above. R represents the energy-absorbing element, i.e., the sound absorption material in the absorption tube of the KU-FRL facility. According to Reference 13,

$$R = \frac{r_A}{1 + \omega^2 r_A^2 C_A^2} \quad \frac{\text{Nsec}}{\text{m}^3} \text{ or rayls} \quad (4.9)$$

where

$$C_A = \frac{V}{\rho c_o^2 bc} \quad \frac{\text{m}^3}{\text{N}} \quad (4.10)$$

Using Reference 14 to estimate the flow resistivity of the absorbent materials, the acoustic resistance of the present absorption tube was initially estimated to be

$$r_A = 800 \text{ rayls}$$

However, correlation of theoretical calculations of Insertion Loss with observed results indicates that the true value is more likely

$$r_A = 150 \text{ rayls}$$

The coupling coefficient, also from Reference 9, is

$$\frac{B_{mn}}{q_r} = \frac{4}{\pi^2} \frac{q}{(m^2 - q^2)} \frac{r}{(n^2 - r^2)} [\cos(q\pi)\cos(m\pi) - 1] [\cos(r\pi)\cos(n\pi) - 1] \quad (4.11)$$

and

$$\alpha(t) = \frac{1}{c_o} \sqrt{\omega_{mn}^2 - \omega^2} \quad (4.12)$$

4.3.3 Prediction of Insertion Loss

Now that panel impedance, cavity impedance, coupling coefficient, and $\alpha(t)$ are known, the Insertion Loss* can be calculated. Insertion Loss is the difference in decibels between the sound pressure level on the excited side of the panel and the transmitted sound level. IL is also discussed in Appendix A.

The derivation of the following relation has been given in Reference 9; only the final result will be repeated here. In general, for normally incident sound,

$$\frac{P_2(x,y,z,t)}{P_1(\text{plane wave})} = \sum_{\substack{q=1,3,5\dots \\ r=1,3,5\dots \\ m=0,2,4\dots \\ n=0,2,4\dots}}^{\infty} KK' \frac{16}{qr\pi^2} \frac{B_{mn} [Z_{mn}(0) \text{cav}] \cosh[(a-x)\alpha(t)]}{[Z_{mn}(0) \text{cav} + Z_{qr} \text{pan}] \cosh[a\alpha(t)]} \cdot \cos\left(\frac{m\pi}{b} y\right) \cos\left(\frac{n\pi}{c} z\right) e^{j\omega t} \quad (4.13)$$

$$\text{where } K = 0.5, \quad m = 0 \\ 1.0, \quad m > 0 \quad (4.14)$$

$$K' = 0.5, \quad n = 0 \\ 1.0, \quad n > 0 \quad (4.15)$$

For comparison with the present experimental results, this equation can be simplified to yield the time average IL at an arbitrary point on the axial centerline of the cavity. The resulting relation is

$$\frac{P_2(x, \frac{b}{2}, \frac{c}{2})}{P_1} = \sum_{\substack{m=0, n=0 \\ q=1, r=1}}^{\infty} KK' \frac{16}{qr\pi^2} \frac{B_{mn} [Z_{mn}(0) \text{cav}] \cosh[(a-x)\alpha(t)] (-1)^{\frac{m+n}{2}}}{[Z_{mn}(0) \text{cav} + Z_{qr} \text{pan}] \cosh[a\alpha(t)]} \quad (4.16)$$

for $m, n = 0, 2, 4, \dots$ and

$q, r = 1, 3, 5, \dots$

*"Insertion Loss IL" as used in this report has the same sense as Noise Reduction NR in Ref. 14.

FREQUENCY ~ Hz

2500

2000

1500

1000

500

0

-500

STYL

SKM

IMPEDANCE

Edge Conditions for
Theoretical Calculations

1,1 Mode: Clamped

All Higher Modes: 50%
Clamped, 50% Simply
Supported

PANEL MODES →

CAVITY
MODES

$Im(0,0)$

$Re(0,0)$

$Im(0,2)$

$Im(0,2)$

$Im(0,0)$

1,1

1,3

3,3

1,5

3,5

5,5

1,7

3,7

5,7

Figure 4.3.3: Impedances of an
18" x 18" x .0387" Aluminum Panel,
and of the KU-FRL Absorption Cavity

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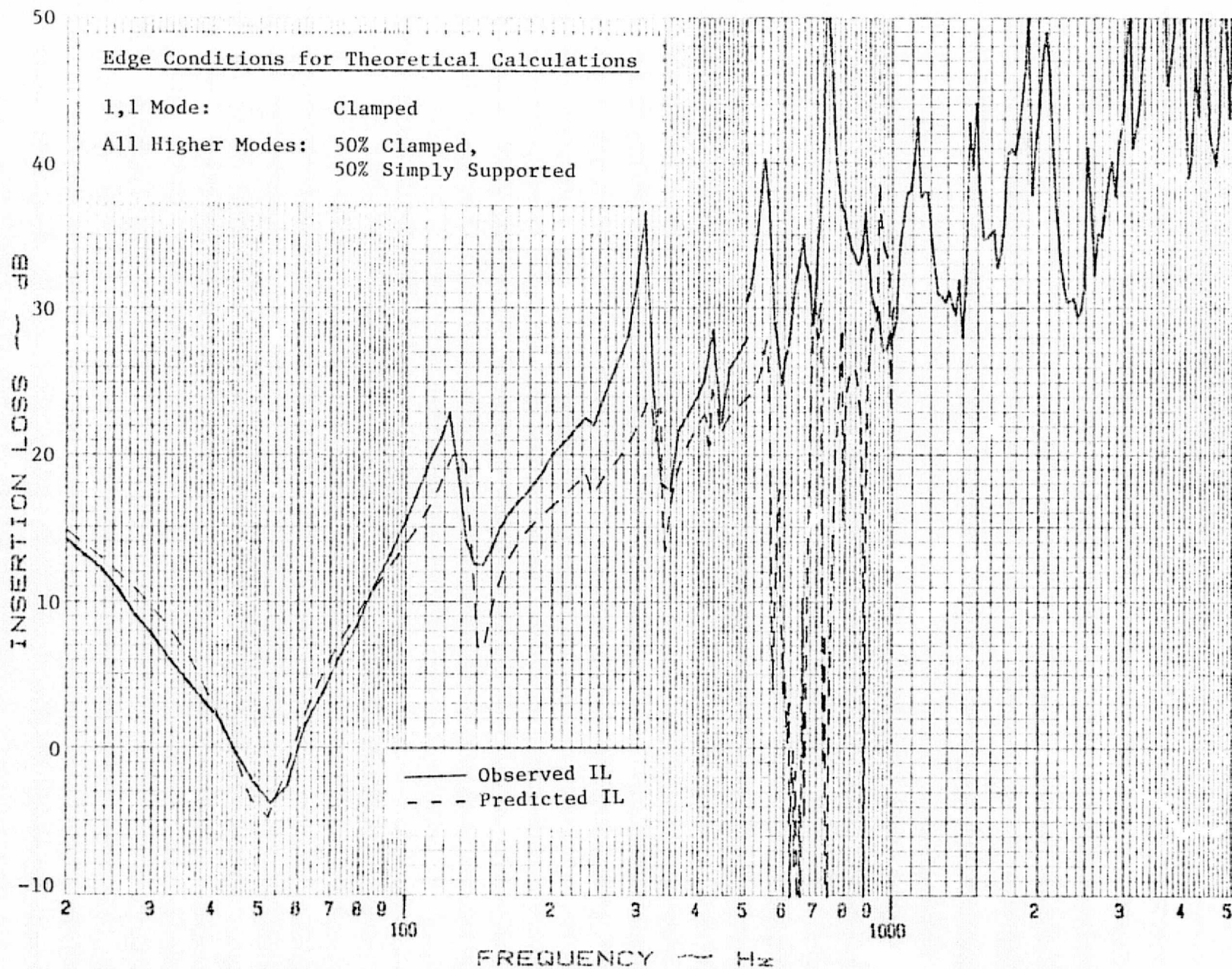
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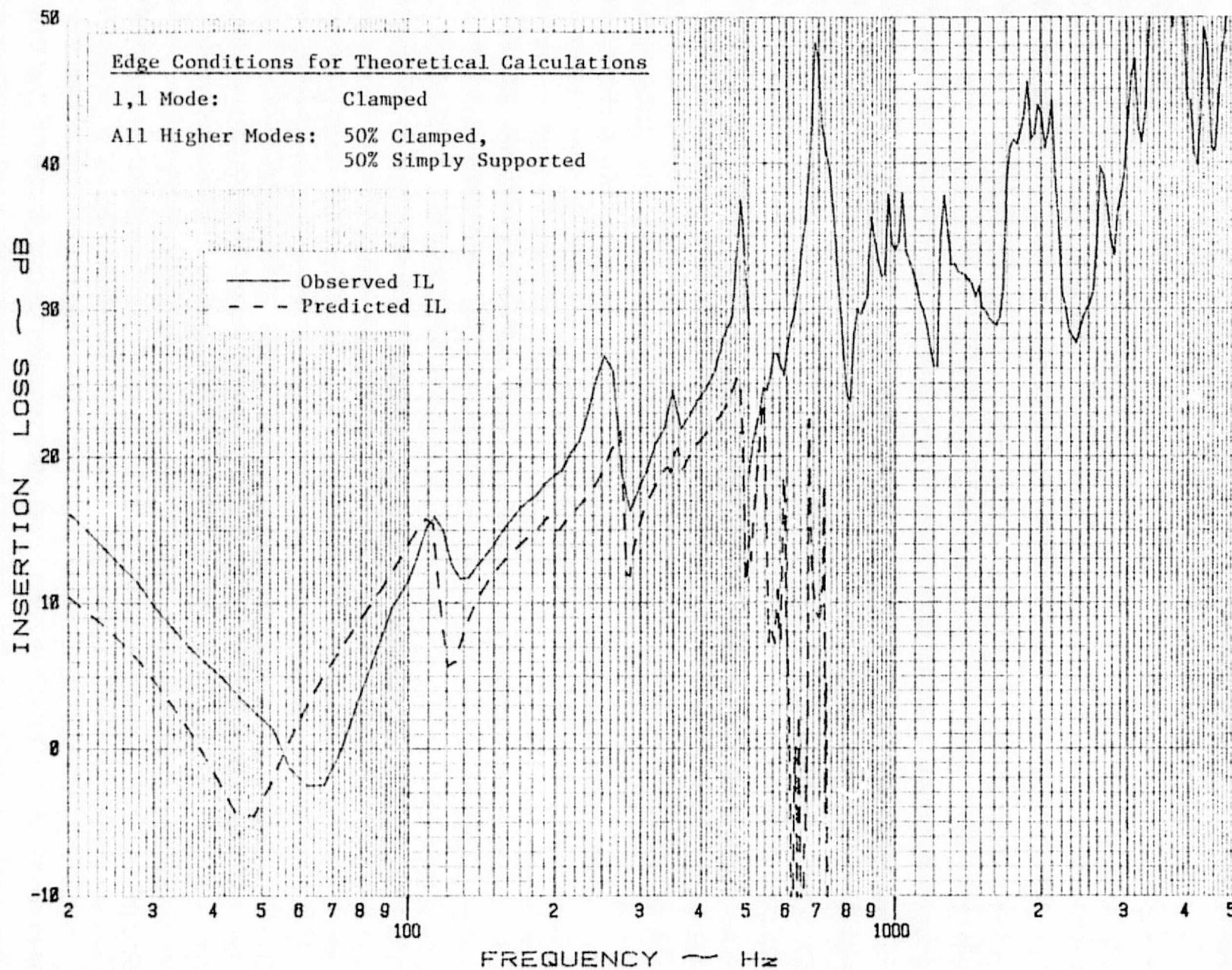
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Figure 4.3.4: Predicted and
Observed Insertion Loss of an 18" x
18" x .0387" Aluminum Panel in the
KU-FRL Test Facility

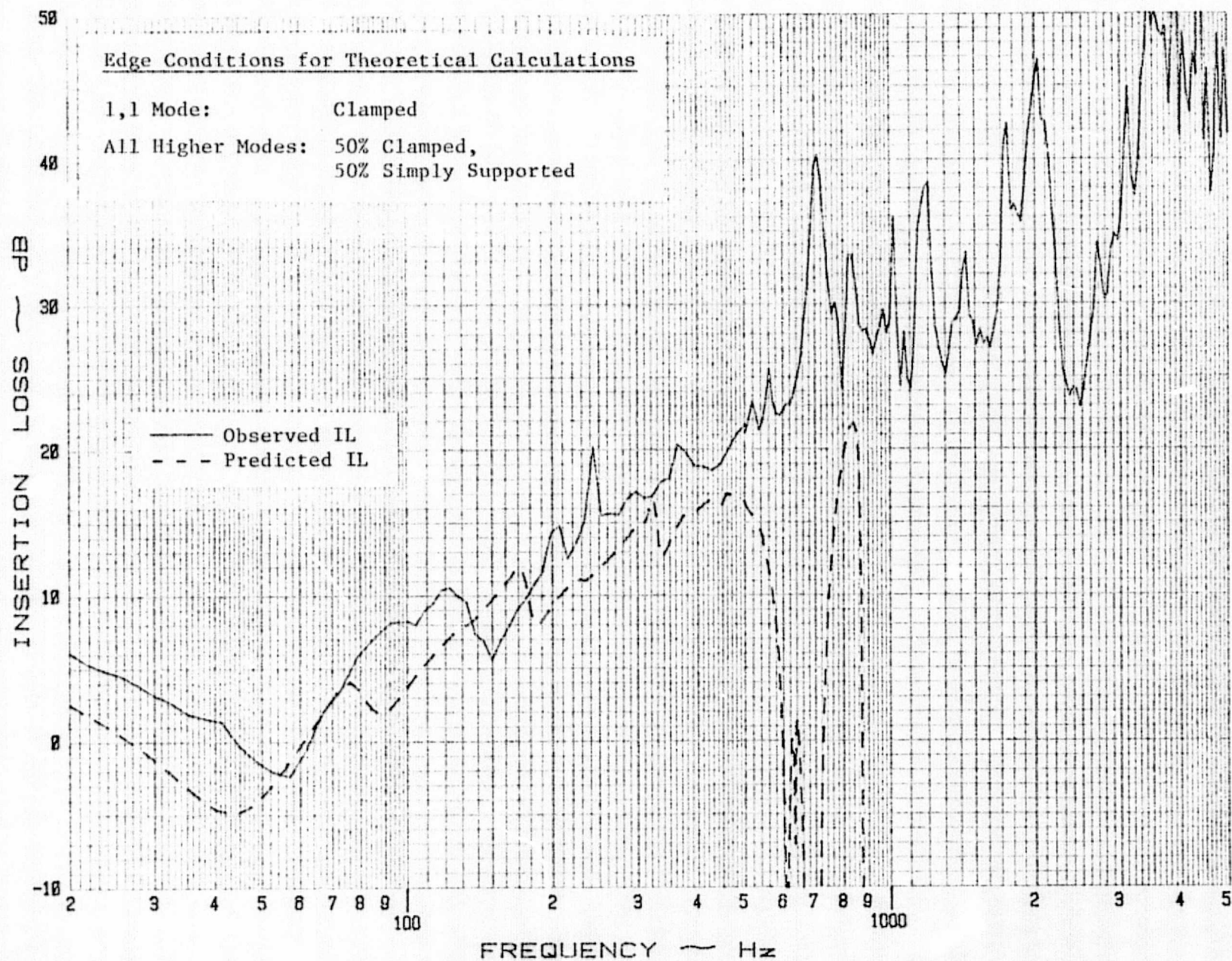
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CALC		REVISED	DATE	Figure 4.3.5: Predicted and Observed Insertion Loss of an 18" x 18" x .032" Aluminum Panel in the KU-FRL Test Facility	UNIVERSITY OF KANSAS PAGE 48
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CALC		REVISED	DATE	Figure 4.3.6: Predicted and Observed Insertion Loss of an 18" x 18" x .020" Aluminum Panel in the KU-FRL Test Facility	UNIVERSITY OF KANSAS PAGE 49
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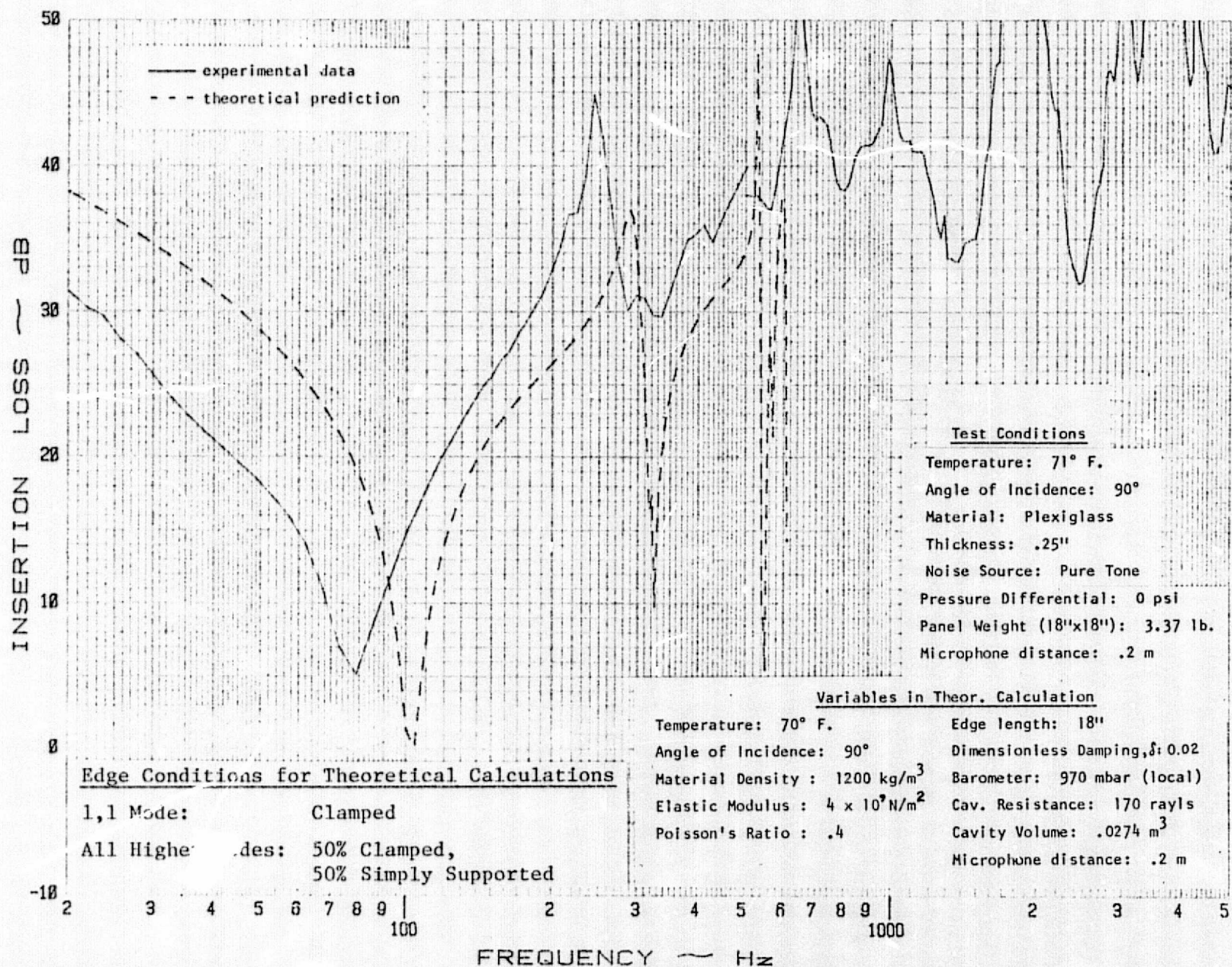


Figure 4.3.7: Predicted and Observed
Insertion Loss of an 18" x 18" x .25"
Plexiglass Panel, in the KU-FRL Faci-
lev.

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$$\text{and} \quad \text{IL}(x, \frac{b}{2}, \frac{c}{2}) = 20 \log \frac{P_2(x, \frac{b}{2}, \frac{c}{2})}{P_1} \quad (4.17)$$

The validity of this theoretical relationship was tentatively verified by successfully repeating the sample calculations of Reference 9. The physical cavity/panel configurations of the sample calculations and experimental observations contained only reactive impedances; that is, all cavity walls were completely reflective. Therefore, all impedance contributions were imaginary.

For comparison with the present experimental results, however, it was necessary to account for energy absorption by the cavity. It was also desirable to include energy absorption (damping) by the test panel. For these cases, the impedances become complex, as detailed above.

A Hewlett-Packard 9825A desktop computer was programmed to sum the contributions of 16 panel modes ($q, r = 1, 3, 5, 7$) and four cavity modes ($m, n = 0, 2$) at discrete frequencies. The results were plotted as Insertion Loss curves.

As indicated in Appendix A, IL can be easily converted to TL. However, for greater sensitivity in comparing calculated and observed data, experimental TL curves were converted to IL curves.

Several data required for IL calculation were not known accurately. Initially, physically reasonable estimates were used for these quantities, and adjustments were made until calculations agreed well with experimental observations. For a clearer picture of the situation, impedance plots such as Figure 4.3.3 were also employed. The quantities iterated in this way were

δ , panel damping

r_A , acoustic resistance of the cavity absorbent material

a , effective cavity length.

Panel damping was found to be negligible, both for rubber and aluminum clamping edges. Cavity acoustic resistance was determined to be about 150 rayls, although there are indications that this figure may be low. Two hundred (200) rayls appears to be the maximum likely value.

The effective cavity length at which reflection occurs was more difficult to determine. It definitely appears to depend on frequency. The following tentative relationship was established:

$$\begin{aligned} a &= 31.172f^{-.8685} & f &\leq 130 \text{ Hz} \\ a &= 1.868 - .2904 \ln(f), & f &> 130 \text{ Hz} \end{aligned} \quad (4.18)$$

These values were obtained by comparing calculated and experimental data for a .0387 in. (.040 in. Nom) thick aluminum panel 18 in. square. The two curves, shown in Figure 4.3.4, agree reasonably well between 20 and 500 Hz.

For other panels, predicted and observed trends agree well as shown in Figures 4.3.5 and 4.3.6. The frequency discrepancies are attributed to incomplete knowledge of panel edge conditions, which has been discussed in the previous section. The primary effect of less rigid panel edge support is to lower the in vacuo natural frequencies. The panel impedance curves (Figure 4.3.3) will then be shifted to the left, and the observed resonance frequencies will also drop. The natural frequencies of the higher modes will probably be shifted by a greater number of Hertz than that of the first mode, as described in the previous section. The effect on IL or TL is not known.

4.3.4 Conclusions and Recommendations

This subsection pertains to the IL prediction method which has just been described.

4.3.4.1 Conclusions

The IL prediction method described above appears to give useful results for plain, unstiffened panels. The prediction accuracy of combined cavity/panel resonance frequencies, and of actual IL levels, appears to depend on the accuracy to which in vacuo panel natural frequencies, i.e. edge conditions, are known. Some of the IL discrepancies above the first resonance may be due to errors in cavity impedance estimation, stemming from inaccurate cavity length or cavity volume assumptions.

Above 500 - 600 Hz, large discrepancies are noted. These appear to be at least partly due to inaccurate cavity length assumptions.

The aim of this prediction method is not to eliminate testing, but to further the understanding and clarify the application of test results. A panel will behave differently in tests than it will in an aircraft fuselage, because the cavity characteristics are different. Knowledge of such behavior can avoid possible costly disappointments when applying test information to flight articles.

At the time this report was being completed, Reference 20 was found to treat much the same problem. A full evaluation of the approach and results of Reference 20 cannot be given, but the following points are noted:

- 1) Both Reference 20 and this report assume that the panel impedance expression of Reference 9 holds for clamped panels. This is probably not true.

- 2) The calculated IL level at resonance of a simple aluminum panel, by the Reference 20 method, disagrees with experiment by 10 dB. The method of this report shows better agreement.
- 3) The use of third octave experimental data in Reference 20 obscures a good deal of information when comparisons are made with calculations. Narrow band data is preferred.

4.3.4.2 Recommendations

It is desirable to eliminate the large discrepancies which occur above 500 - 600 Hz. The cavity impedance calculations are incorrect at these frequencies, probably due to

- 1) incorrect estimates of effective cavity length;
- 2) inappropriate determination of acoustic capacitance C_A used to find real cavity impedance.

The length estimate needs to be improved, possibly by more closely correlating it to physical realities including its maximum possible length. The persistent high TL peaks at about 730 Hz and two or three higher frequencies may also provide clues. Real cavity impedance should be found by using a capacitance value determined from cavity reactance. The method presently used assumes that the wavelength is much larger than the cavity dimensions. This assumption is only valid up to about 500 Hz.

Another assumption needs to be checked. The panel impedance calculation, Eq. 4.2, was given in Reference 9 for simply supported plates. For expedience, this relation has been assumed to hold for clamped plates as well. A theoretical investigation is recommended to find a more accurate expression.

4.4 Large Deflections

In most theoretical treatments of plate vibration and acoustic radiation, vibrational displacements are assumed to be "small." This assumption simplifies the analytical work. For practical use a small "deflection" may be defined as a displacement less than $1/3$ the plate thickness.

The test results of the present research have been compared to theoretical predictions. In all cases the theories have assumed small deflections. The question has arisen as to the actual magnitude of panel displacements during tests.

A brief investigation was conducted at the time of the original calibration (Reference 2). Results were inconclusive due to excessive weight of the accelerometer used, and electrical noise in the measuring circuit. It is planned to repeat this investigation in the next phase of the project. A miniature accelerometer and a low-noise circuit will be used.

It may be useful to see what the results of large deflections would be, according to theory. Reference 7 indicates that the natural frequency will increase by an amount that depends on the edge conditions and the amplitude/thickness ratio. This data is summarized in Figures 4.4.1 and 4.4.2.

Panel impedances may also be affected by the occurrence of large deflections. This effect would probably be best studied experimentally, as analysis would probably be quite difficult to carry out.

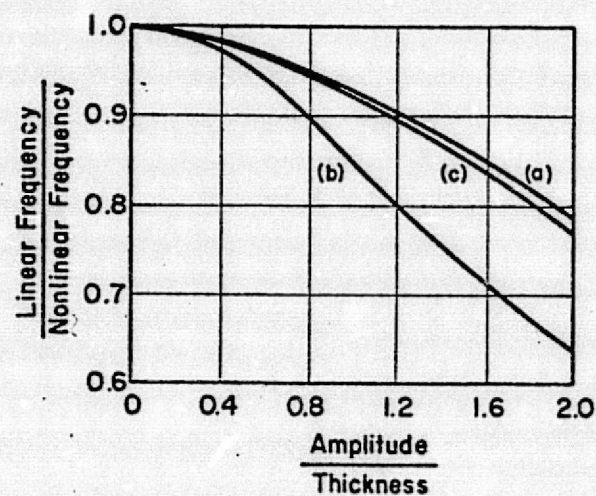


Figure 4.4.1: Effect of Large Amplitude on the Lowest Natural Frequency of a Clamped Square Plate for Three Cases of Inplane Restraint; $\nu = 0.3$. From Ref. 7

FOR BOTH FIGURES	
(a) =	All Edges Stress-free
(b) =	All Edges Immovable
(c) =	All Edges Movable

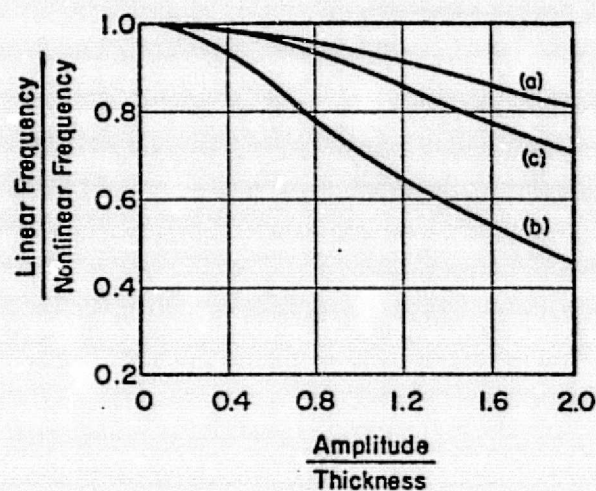


Figure 4.4.2: Effect of Large Amplitude on the Lowest Natural Frequency of a Simply Supported Square Plate for Three Cases of Inplane Restraint; $\nu = 0.3$. From Ref. 7

CHAPTER 5

SOUND TRANSMISSION THROUGH PANELS WITH HOLES

This chapter contains a discussion of a method of predicting the transmission loss of panels having holes in them. The TL curves predicted for sample panels are compared with experimental values for the same panels.

5.1 Theory

In architectural acoustics, the total sound pressure transmitted through a barrier is considered equal to the sum of the sound pressures transmitted through each distinct region of the barrier. (Ref. 14)

$$P_{trans} = P_{inc} \sum \tau_i S_i \quad (5.1)$$

where

P_{trans} = total transmitted sound pressure

P_{inc} = incident sound pressure

τ_i = transmission coefficient of region i

S_i = area of region i

This concept can be applied to a panel with a hole. The hole constitutes a region of practically perfect sound transmission, and the remainder of the panel another region having a known (or calculable) transmission coefficient.

τ_1 = panel transmission coefficient

$$= 10^{\frac{-(TL)_{solid\ panel}}{10}}$$

τ_2 = hole transmission coefficient

$$= 1.0$$

S_1 = panel area, excluding hole

S_2 = hole area

The transmission of a panel with a hole is

$$\begin{aligned} TL &= 10 \log_{10} \frac{S_1 + S_2}{\tau_1 S_1 + \tau_2 S_2} \\ &= 10 \log_{10} \frac{S_1 + S_2}{\tau_1 S_1 + S_2} \end{aligned} \quad (5.2)$$

The upper limit of the TL of a panel with a hole can be found by assuming that the non-hole panel area has an infinite TL:

$$\begin{aligned} \tau_1 &= 0 \\ TL_{\max} &= 10 \log_{10} \frac{S_1 + S_2}{S_2} \end{aligned} \quad (5.3)$$

This theory considers only the hole area relative to the panel area and the TL of the solid panel. It does not account for changes in plate vibrational behavior, or for changes in panel or cavity (room) impedances, which are likely to occur.

The likely changes in panel resonance frequency can be estimated from Reference 7. With a circular central hole, the following trend can be expected.

<u>Hole radius</u> <u>plate edge length</u>	<u>Hole Area</u>	<u>Resonance frequency shift</u>
0	0	0
0.1	0.8%	-1.6%
0.2	3.1%	-1.4%
0.3	7.1%	+5.2%

This effect appears slight.

The changes in effective impedances are more difficult to determine and will not be estimated here.

5.2 Comparison of Predicted and Measured Transmission Loss

A number of tests were made to check the validity of the prediction method described above. A square aluminum panel .040 in. thick was first tested, with a speaker-panel spacing of one inch. The effective panel area was 18 inches square, and the edges were considered to be clamped. The same panel was then successively retested with a progressively larger hole in the center. All holes were round. The resulting TL curves can be found in Appendix B.

Holes smaller than 1/2 inch diameter, i.e. .06% panel area, reduced the transmission loss very little. Larger holes had greater effect.

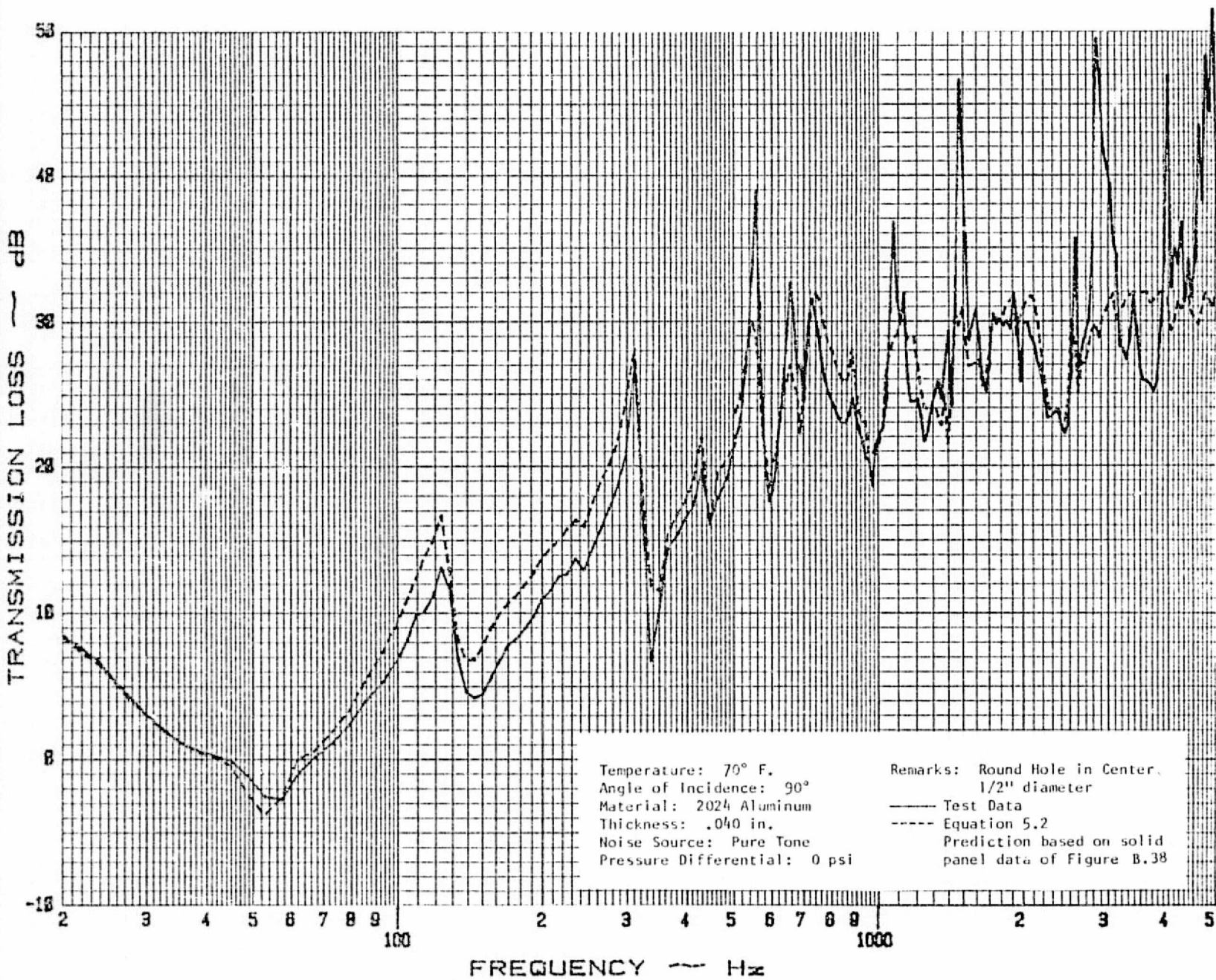
The transmission loss curve (Figure B.34) of the intact panel--before any holes were made--was used as the starting point to check Equation 5.2. At each data point, the transmission coefficient τ_1 was found from the TL value, and used in Equation 5.2 to predict the TL of a panel with a hole. This was done on the HP 9825A desktop computer system for several hole sizes. In Figures 5.1 to 5.4, these predicted TL curves are compared with corresponding experimental curves.

Equation 5.2 appears quite accurate for the panel with the 1/2 inch hole, and less accurate for larger holes. The greatest average error occurs in a frequency band up to 200 Hz wide, straddling the second major resonance (valley) frequency. Errors in this range may be due to the "line of sight" sound path from the center speaker, through the hole to the microphone.

At higher frequencies, the predicted curve levels off, as Equation 5.3 suggests. It is not certain why the experimental curves continue

to show large fluctuations. However, the regular frequencies of the peaks and valleys suggest the presence of cavity modes (standing waves) connected with the 18 inch distance between horizontal and vertical walls in the region of the receiving microphone. (This is especially noticeable in Figure 5.4.) If the extreme peaks and valleys are ignored, however, then the predicted values coincide well with the mean experimental data.

The Transmission Loss of panels with holes or slits could be more rigorously calculated by using an impedance formulation similar to that of Section 4.3. (See Chapter 5 of Reference 13). Such a calculation method would most likely be more accurate than the one shown here; but certain quantities, e.g. impedance of a cavity with a hole in a vibrating wall, could be difficult to determine.



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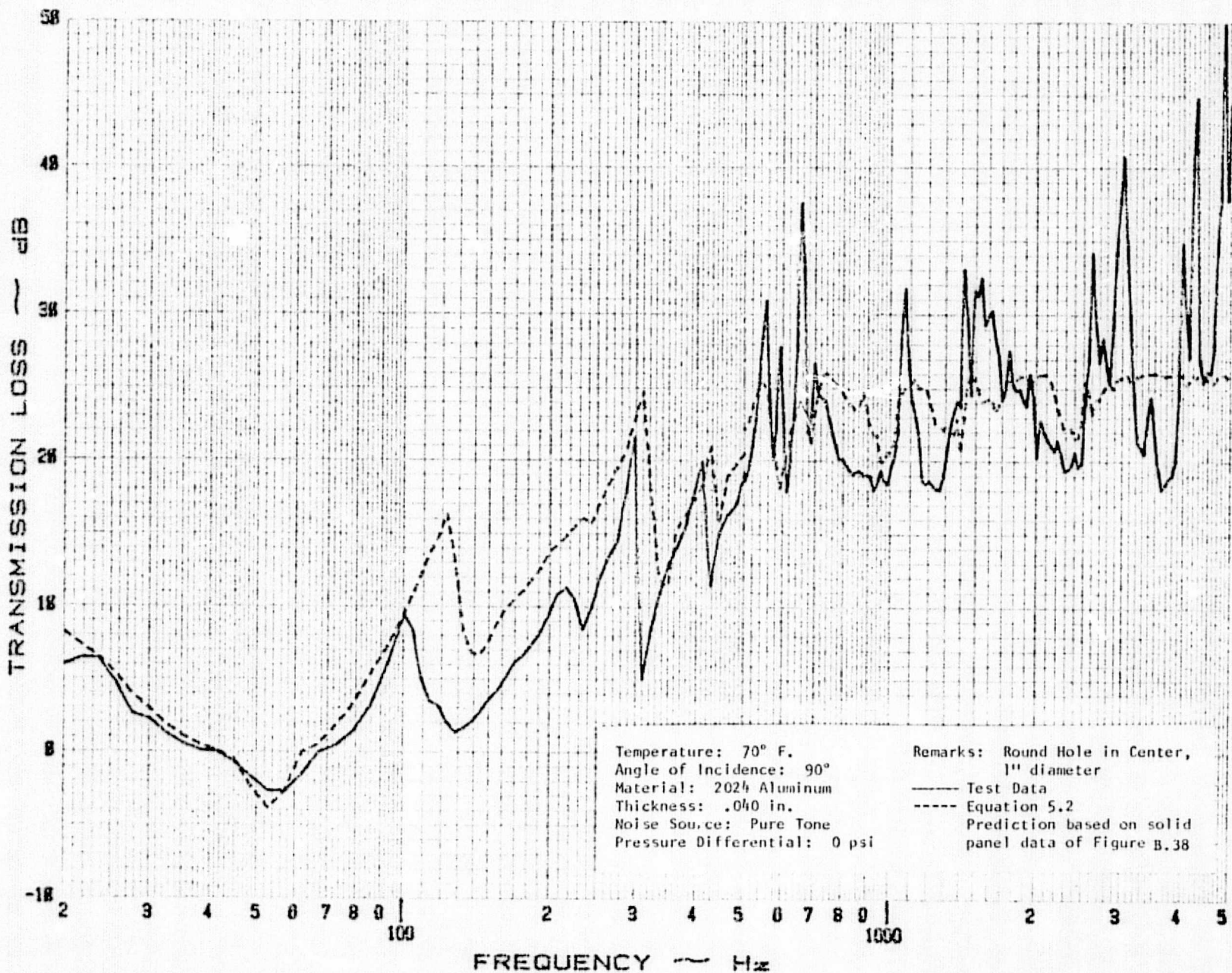
Figure 5.1: Predicted and Measured
Sound Transmission Loss of a Square
Aluminum Panel with a .061% Hole

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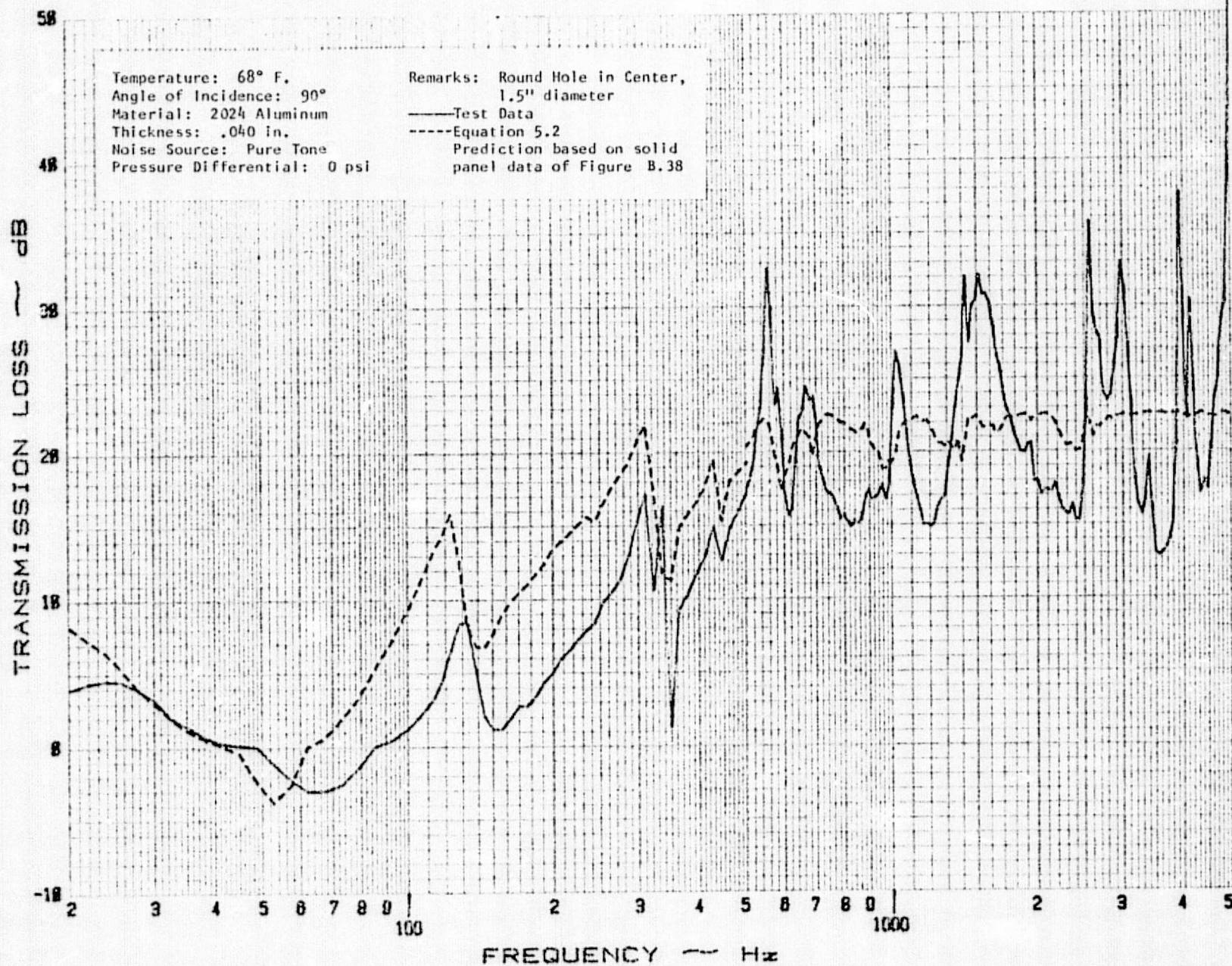


Figure 5.3: Predicted and Measured
 Sound Transmission Loss of a Square
 Aluminum Panel with a .55% Hole

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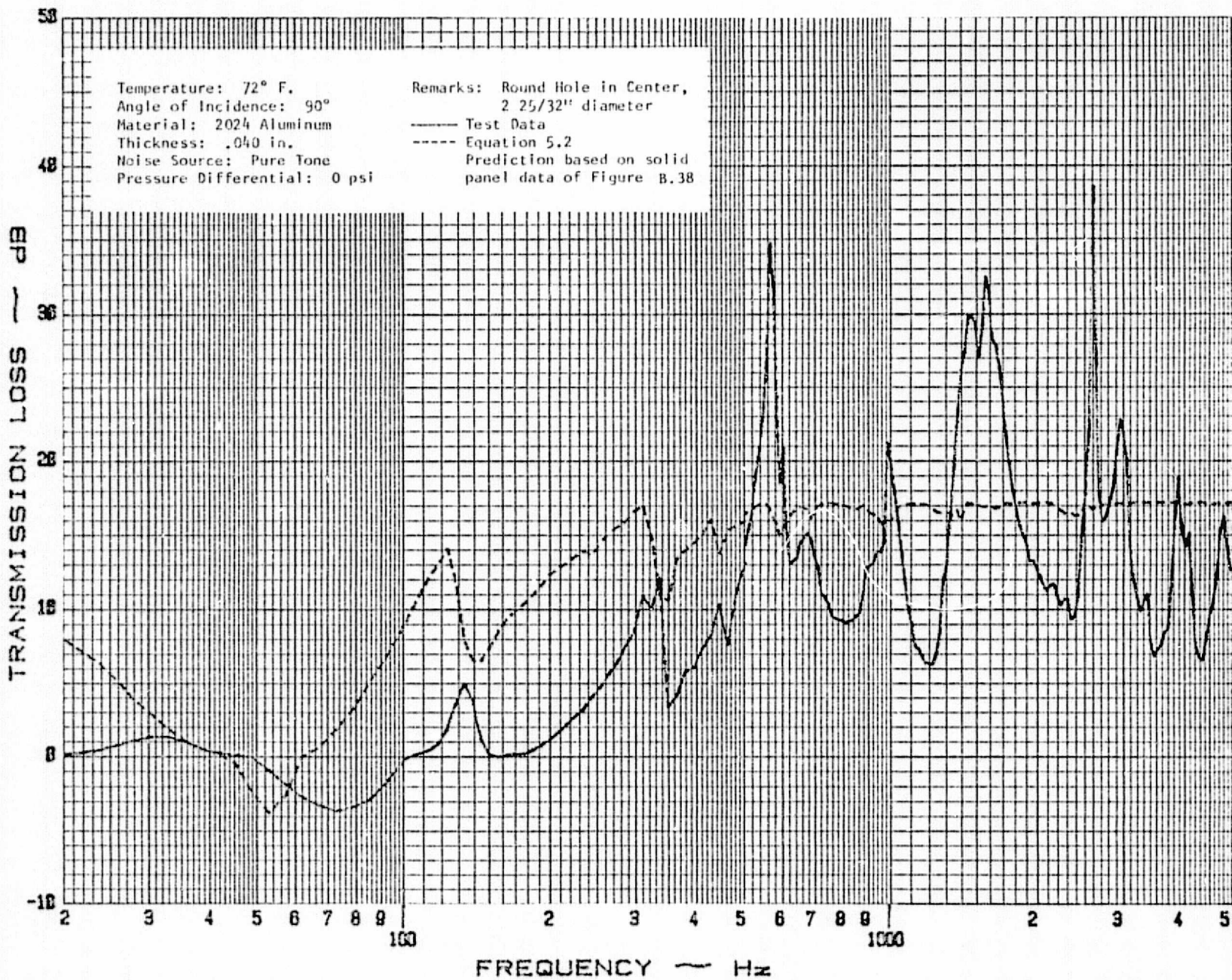


Figure 5.4: Predicted and Measured
 Sound Transmission Loss of a Square
 Aluminum Panel with a 1.88% Hole

CHAPTER 6

POTENTIAL OF TUNED DAMPERS

For many years, attempts have been made to reduce structural vibration. One of the more novel approaches has been the so-called tuned damper, sometimes called a vibration absorber.

This device consists of a small mass attached by a spring to the vibrating structure. If the natural frequency of the attached spring-mass system matches the natural or excitation frequency of the original system, the magnitude of vibration of the latter will be reduced significantly. Excellent vibration attenuation is possible; but natural frequencies must be matched, or tuned, quite accurately. This requirement generally rules out the use of tuned dampers in mass produced goods. The care required for accurate tuning cannot be economically justified. See References 15 and 16.

The requirement for accurate tuning can be relaxed somewhat if a true damper is added to the spring-mass system. As an illustration of this effect, consider the system of Figure 6.1, whose equations of motion are

$$M_1 \ddot{x}_1 = -k_1 x_1 - c_1 \dot{x}_1 + k_2 (x_2 - x_1) + c_2 (\dot{x}_2 - \dot{x}_1) + P \sin \omega t \quad (6.1)$$

$$M_2 \ddot{x}_2 = -k_2 (x_2 - x_1) - c_2 (\dot{x}_2 - \dot{x}_1) \quad (6.2)$$

The transfer function of structural displacement, or for the present purposes panel displacement, is

$$\frac{x_1(s)}{P(s)} = \frac{k_2 + c_2 s + M_2 s^2}{(k_1 + c_1 s + M_1 s^2)(k_2 + c_2 s + M_2 s^2) + M_2 k_2 s^2 + M_2 c_2 s^3} \quad (6.3)$$

This can be restated in the frequency domain by substituting $s = j\omega$.

In that form the transfer function was programmed for solution on a

Hewlett-Packard 9825A calculator.

The goal was to define a specific system that could be tested in the KU-FRL as a proof of concept. An aluminum panel 18 inches square comprises the vibrating structure. Assuming that 20% of the panel surface is effective in 1,1 mode vibration, the effective mass $M_1 = .00054 \text{ lb sec}^2/\text{in}$. From considerations of the observed resonance frequency of such a panel in this facility, the effective panel stiffness k_1 was estimated to be approximately 60 lb/in. The sum of internal panel damping and edge damping was estimated to be .00159 lb sec/in. This was based on weakly damped panel vibration in vacuo, with $\zeta = .007$.

The spring and damper elements can be combined in one unit, if an elastomeric commercial equipment mount is used. The Barry Model 5820-2 appears potentially useful. It weighs 0.15 oz (4.5 g) and is available in natural rubber, neoprene, and Barry Hi-dampTM silicone elastomer compositions. Under dynamic conditions its stiffness k_2 is 19 lb/in.

The following mass-mount combinations appear to be potentially effective in reducing the vibration amplitude of an 18 in. square panel under present KU-FRL test conditions.

<u>Mass</u>	<u>Mount</u>
.0003 lb sec ² /in	Barry 5820-2,
or .116 lb _m	natural rubber or neoprene
or 52.5 g	($\zeta = .05$) (Ref. 17)
.0004 lb sec ² /in	Barry 5820-2,
or .154 lb _m	Hi-damp TM
or 70.0 g	($\zeta = .15$) (Ref. 17)

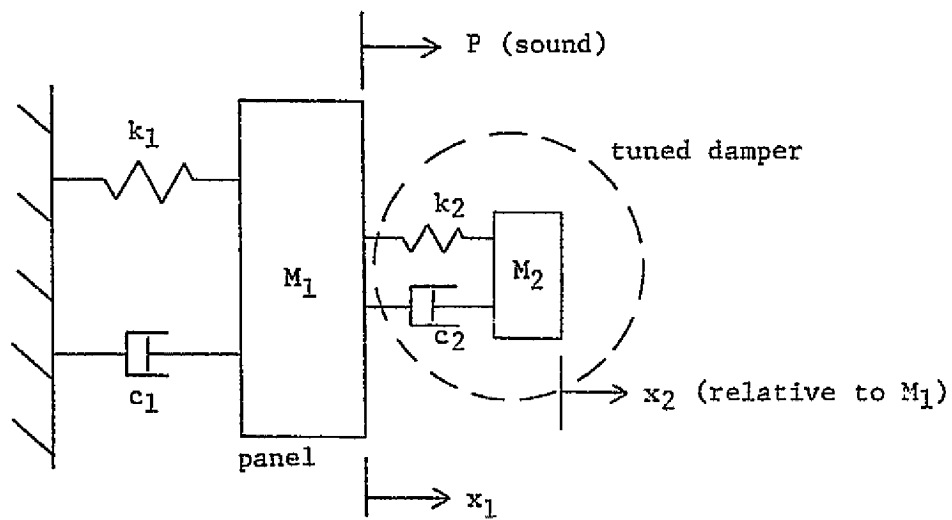


Figure 6.1: Arrangement of a Tuned Damper

Figure 6.2 compares the vibration magnitude term $x_1(\max)/P$ of panels with and without these tuned damper treatments. For an untreated panel $x_1(\max)/P$ is very large at resonance. The first combination reduces vibration significantly, and the second combination appears even more effective.

One possible drawback to the use of tuned dampers is their cooling requirement. Reference 18 indicates that problems have been encountered in practice when tuned dampers have to absorb significant energy. If the resulting heat is not eliminated, degradation of performance, and physical disintegration are quite possible.

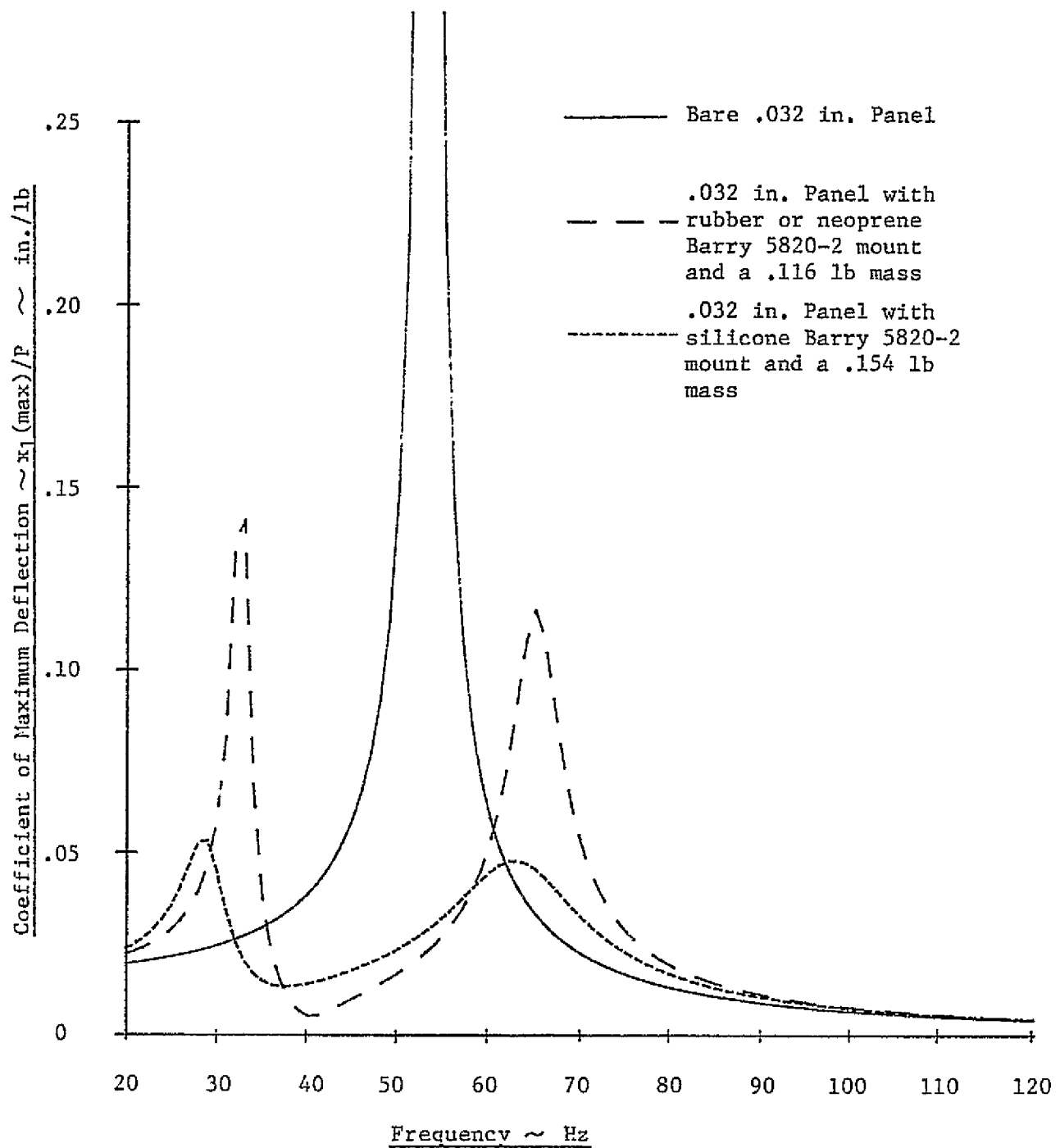


Figure 6.2: Calculated Displacement Amplitude of a Vibrating Square Aluminum Plate

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

7.1 Conclusions

Up to the present time the NASA-KU-Industry Noise Research Program has achieved the following main goals:

- Accumulation of an extensive data base of
flat panel transmission loss;
- Construction and preparation of equipment for
panel transmission loss tests at non-normal
(slanted) sound incidence;
- Enhanced understanding of cavity influences on
test data, as demonstrated by a comprehensive
theory-based prediction method.

A large number of tests have been conducted on square flat panels at normal sound incidence. These tests have consumed more time than originally estimated, because of necessary revisions in testing procedures. Acquisition of a dedicated desktop computer in the follow-on phase is expected to reduce total testing time by at least half. Most of the tests were conducted on panels which appeared both practical and potentially effective in reducing noise transmission. The intention has been to maximize the usefulness of project efforts.

The special test sections for slanted (non-normal incidence) panel testing have been designed and built. They will also permit testing of flat panels at normal incidence and, with a support plate, curved panel testing. The four special test sections are presently undergoing final preparations for operation.

The improved understanding of cavity influences has significantly aided progress in the Noise Research Program. The prediction method in its present form shows good agreement with experimental observations in the three most important regions of panel vibration: the stiffness-controlled region, the resonance-controlled region, and the mass-law region. This has served to explain some unexpected experimental results, namely negative transmission loss and peaks in the TL curves. This insight into cavity effects is expected to facilitate the intelligent application of test data to successful aircraft noise reduction.

Another point of progress has been the detail design of the uni-axial test device. Mechanical compatibility with the existing test facility has been achieved. Construction has been delayed primarily by the need to attend to other more urgent work, and by considerations of later extending capabilities to bi-axial testing. The device is expected to be completed in March 1978.

7.2 Recommendations

There are three areas of test facility work that need attention. Clamping conditions on all test sections need to be improved. The present conditions have provided very useful comparative data, but are insufficient for determination of cavity characteristics by correlation with theory. Unless the edges are clamped, their conditions cannot be known with any certainty. Work should also continue on the panel-tensioning device so that it can become operational soon. Additionally, a short investigation should be made to locate a new microphone position which gives average TL closer to mass-law. This appears to eliminate the main TL-raising effect of cavity resonances much more easily than could a correction in data reduction, because the affected frequencies

vary slightly from test to test.

Theoretical work directed at improvement of the prediction method would be very helpful. This work should probably take the form of improved cavity length estimates and development of an expression for clamped panel impedance. These improvements are expected to result in very good predictive accuracy over the entire frequency range of interest, 20 to 5000 Hz. The cavity characteristics of the special test sections also need to be determined. If possible, this should proceed simultaneously with the work on the original absorption cavity characteristics, in order to permit timely interpretation of slanted panel test data.

At this point in the Noise Research Program, it may be worthwhile to consider how hindsight would solve some of the problems which have been encountered.

All uncertainties in clamping could have been avoided by clamping a 2-inch-wide strip along the edges of all test panels by means of a through-bolting arrangement, or a similar "overkill" method. It is difficult, indeed, to really clamp a test panel; and this potential source of concern is best dealt with early.

Cavity resonances fall into the same category. It is best to design them out in the beginning. Three possibilities exist. The cavity can be made small enough that its longest dimension is less than the wavelength of the highest frequency of interest. Or, the cavity walls could be arranged so that no two are parallel. In this case a small amount of absorbing material on the walls may easily eliminate oblique and tangential standing waves up to a sufficiently high frequency even in a relatively large cavity, e.g. 0.5 m x 0.5 m

x 0.5 m. The third possibility is using an open air termination, or approximating it with a good anechoic chamber. The first and third possibilities are probably the most effective as well as the most difficult. Selection of the preferred approach would best be based on a determination of preferable and tolerable cavity impedances.

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APPENDIX A

ENERGY DISSIPATION CORRECTION FOR

PANELS POSSESSING DAMPING, AND CORRECTIONS

USED IN DATA REDUCTION

APPENDIX A

CORRECTIONS APPLIED TO TEST DATA

This Appendix describes three corrections which have been applied to test data taken up to the present. The purpose of these corrections is to eliminate extraneous effects which could cause difficulties in applying the data to practical problems.

A.1 Reflection

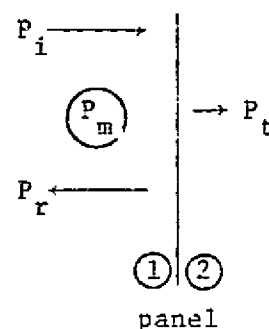
This correction converts Insertion Loss, IL, which is measured, to Transmission Loss, TL, desired. Insertion Loss is the difference in dB between measured pressure P_m on the source side, and transmitted pressure P_t on the receiver side. Transmission Loss is the difference in dB between incident sound pressure and transmitted sound pressure.

P_i = incident sound pressure

P_r = reflected sound pressure

P_m = sound pressure measured on source (speaker) side of panel

P_t = transmitted sound pressure (measured behind panel)



if $(\rho c)_1 = (\rho c)_2$

then the acoustic power relationship is:

$$I_i = I_r + I_t \quad (A.1)$$

$$P_i^2 = P_r^2 + P_t^2 \quad (A.2)$$

$$P_r^2 = P_i^2 - P_t^2 \quad (A.3)$$

$$P_m = P_i + P_r \quad (A.4)$$

$$\begin{aligned}
 P_i^2 &= (P_m - P_t)^2 + P_t^2 \\
 &= P_m^2 - 2P_m P_t + P_t^2 + P_t^2
 \end{aligned}
 \tag{A.5}$$

$$P_i = \frac{1}{2} \left(P_m + \frac{P_t^2}{P_m} \right)
 \tag{A.6}$$

$$\begin{aligned}
 TL &= 20 \log_{10} \frac{P_i}{P_t} \\
 &= 20 \log_{10} \frac{1}{2} \left(\frac{P_m}{P_t} + \frac{P_t}{P_m} \right)
 \end{aligned}
 \tag{A.7}$$

$$\begin{aligned}
 IL &= SPL_m - SPL_t \\
 &= 20 \log_{10} \frac{P_m}{P_t}
 \end{aligned}
 \tag{A.8}$$

$$TL = 20 \log_{10} \left(10^{IL/20} + 10^{-IL/20} \right) - 6$$

(A.9)

This is the equation used to calculate transmission loss at each frequency. The insertion loss IL is the difference between source side and transmitted SPL's. Figure A.1 compares IL and TL data.

A.2 Pressurized Panels

This correction accounts for acoustic impedance differences due to pressure differences on each side of a panel. This correction is from Reference 19. Acoustic velocity is:

$$\begin{aligned}
 c &= \sqrt{\gamma RT} \\
 \text{and air density is:} \\
 \rho &= \text{const} \left(\frac{P}{P_1} \frac{T_1}{T} \right)
 \end{aligned}$$
(A.10)

Assume $T_1 = T_2$.

The correction to IL for a pressure difference ΔP is

$$\begin{aligned}
 C_{\Delta P} &= 20 \log_{10} \frac{\rho_2 C_2}{\rho_1 C_1} \\
 &= 20 \log_{10} \frac{P_2}{P_1} \\
 &= 20 \log_{10} \left(1 - \frac{\Delta P}{P_1} \right)
 \end{aligned} \tag{A.11}$$

This correction is added to the measured IL. This correction was rarely observed to affect TL by more than 2 dB. In most cases the change was insignificant.

A.3 Energy Dissipation (Damped Panels)

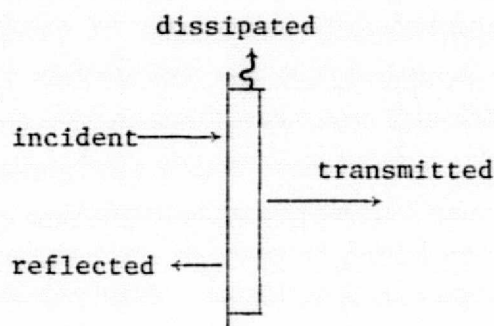
From Section A.1, for a bare panel,

$$P_i = \frac{1}{2} \left(P_m + \frac{P_t^2}{P_m} \right) \tag{A.12}$$

A bare panel is used to obtain P_i

and

$$\begin{aligned}
 \text{SPL}_i &= 20 \log_{10} \left(\frac{P_i}{2 \times 10^{-5} \text{ N/m}^2} \right) \\
 &= 20 \log_{10} \left(10^{\frac{\text{SPL}_m}{20}} + 10^{\frac{2(\text{SPL}_t) - \text{SPL}_m}{20}} \right)
 \end{aligned} \tag{A.13}$$



Then, without changing the amplifier power settings, the bare panel is replaced by the damped panel to be tested, and the transmitted sound is measured:

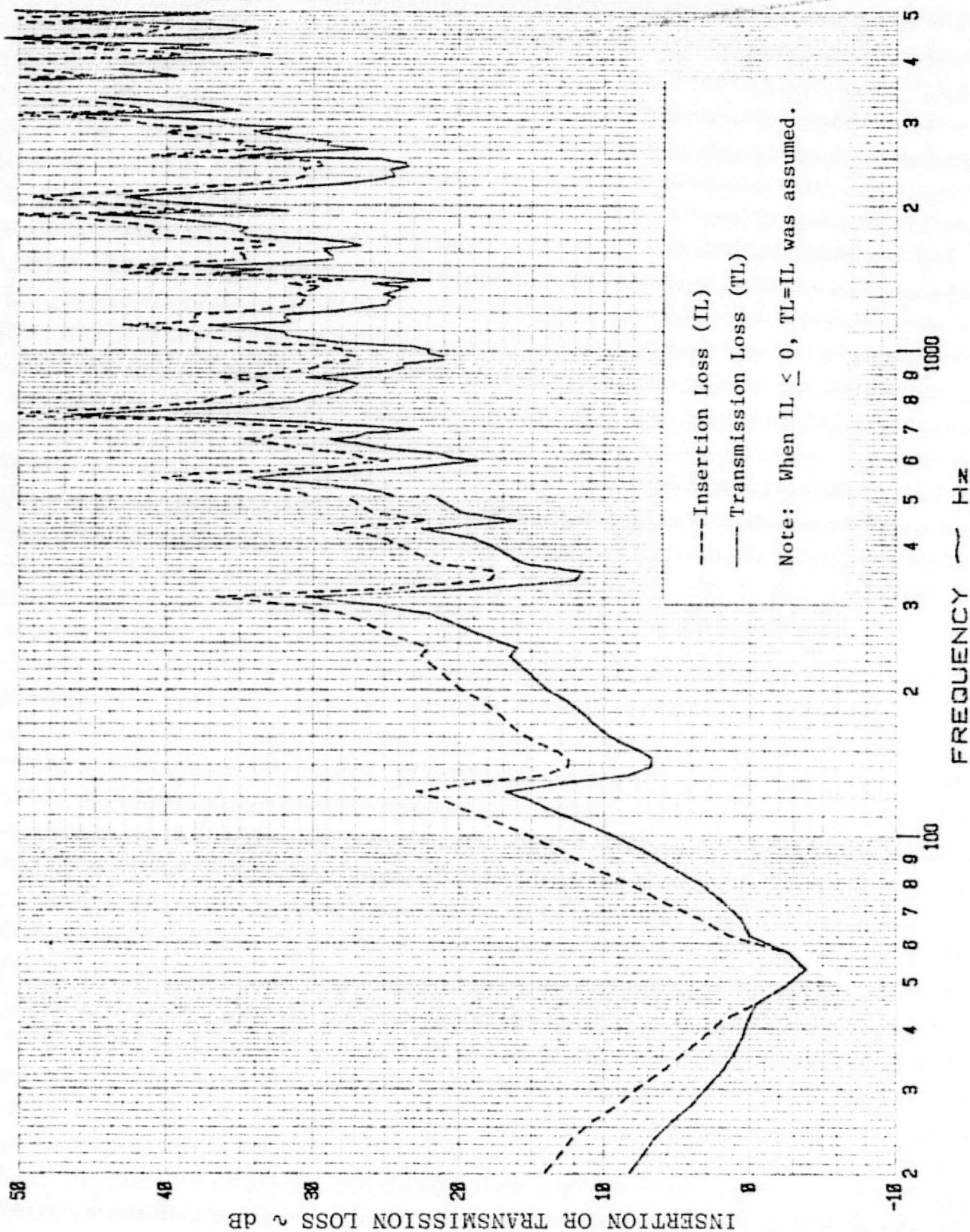
$$\text{SPL}_{td} = \text{SPL transmitted by damped panel}$$

The TL of the damped panel is

$$\text{TL}_d = \text{SPL}_i - \text{SPL}_{td}$$

where SPL_i has been obtained from the bare panel above.

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CALC			REVISED	DATE	Figure A.1: Comparison of Experimental IL and TL Data for an Aluminum Panel 18" x 18" x .0387"	91
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APPENDIX B

PANEL TEST DATA

APPENDIX B

PANEL TEST DATA

Tests on panels listed in Table B.1 have been carried out up to mid-January, 1978. These tests were carefully performed, and are considered useful for purposes of qualitative comparison. Figures referred to in this Table can be found on the pages following. Numbers in parentheses refer to manufacturers of acoustic materials. See Table B.2.

All the tests were performed at temperatures between 67° and 73°F. The average local atmospheric pressure was about 970 mbar.

Table B.1 List of Figures

<u>Figure</u>	<u>Material</u>	<u>Thickness (in.)</u>	<u>Test Condition or Treatment</u>
<u>Damping Treatments</u>			
B.1	Aluminum	.025	A-13, (1) No DC*
B.2	Aluminum	.025	A-13, double thick (1) No DC
B.3	Aluminum	.025	TNB-102 w/PSA, (1) No DC
B.4	Aluminum	.025	Sound Stopper Paste (2) No DC
B.5	Alum. Sandwich	2 x .016	IC-998, recom. appl. (3) No DC
B.6	Aluminum	.032	bare (compare with 2 x .016)
B.7	Aluminum	.025	A-13, (1) with DC
B.8	Aluminum	.025	A-13, double thick (1), with DC
B.9	Aluminum	.025	TNB-102 w/PSA (1), with DC
B.10	Aluminum	.025	Sound Stopper paste (2) with DC
B.11	Alum. Sandwich	2 x .016	IC-998, recom. appl. (3) with DC
B.12	Alum. Sandwich	2 x .016	IC-998, heavy appl. (3) with DC
B.13	Aluminum	.032	TNB 101 w/foam & PSA (1) with DC
B.14	Aluminum	.040	TNB 101 w/foam, A-13 & PSA (1) with DC
B.15	Aluminum	.032	Y-370, 40% cov. (4) with DC
B.16	Aluminum	.032	Y-370, 70% cov. (4) with DC
B.17	Aluminum	.032	Y-370, 100% cov. (4) with DC
B.18	Aluminum	.025	Y-370 and 4 stiffeners, with DC
B.19	Alum. Honeycomb	.50	Y-370 (4) with DC

* DC = Dissipation Correction (See Appendix A).

Table B.1 (continued)

<u>Figure</u>	<u>Material</u>	<u>Thickness (in.)</u>	<u>Test Condition or Treatment</u>
<u>Foam and Batting (all with DC)</u>			
B.20	Aluminum	.032	Pyrell 4 lb. foam (5)
B.21	Aluminum	.020	Fiberglass batt, 1" (6)
B.22	Aluminum	.020	Fiberglass 6 lb/ft ³ (7)
B.23	Aluminum	.020	Fiberglass 3½ lb/ft ³ (7)
<u>Stiffened (No DC)</u>			
B.24	Aluminum	.025	Two intersecting, joined stiff'rs (10)
B.25	Aluminum	.025	Two overlapping perpendicular stiffeners (10)
<u>Honeycombs</u>			
B.26	Aluminum	.032	.50" paper honeycomb (8), No DC
B.27	Aluminum	.032	.50" paper honeycomb (8) with DC
B.28	Aluminum	.032	.50" pap. hc. w/constr. layer, with DC
B.29	FFAC* Honeycomb	.125	1/8" cells (8) with DC
B.30	FFAC Honeycomb	.125	1/8" cells, A-wt fiberglass (8) with DC
B.31	FFAC Honeycomb	.25	1/8" cells (8) with DC
B.32	FFAC Honeycomb	.50	1/8" cells (8) with DC
B.33	FFAC Honeycomb	.50	1/4" cells (8) with DC
B.34	FFAC Honeycomb	.50	glass wool in 1/4" cells (8) w/DC
B.35	FFAC Honeycomb	1.00	1/8" cells (8) with DC
B.36	FFNC* Honeycomb	.50	1/8" cells (8) with DC
B.37	FFNC Honeycomb	1.00	1/8" cells (8) with DC
<u>Holes and Slits (no DC)</u>			
B.38	Aluminum	.040	reference solid panel
B.39	Aluminum	.040	1/16" dia. hole
B.40	Aluminum	.040	3/32" dia. hole
B.41	Aluminum	.040	1/8" dia. hole
B.42	Aluminum	.040	11/64" dia. hole
B.43	Aluminum	.040	15/64" dia. hole
B.44	Aluminum	.040	3/8" dia. hole
B.45	Aluminum	.040	1/2" dia. hole
B.46	Aluminum	.040	3/4" dia. hole
B.47	Aluminum	.040	1" dia. hole
B.48	Aluminum	.040	1½" dia. hole
B.49	Aluminum	.040	2 25/32" dia. hole

*Fiberglass Faces, Aluminum Core. Faces are C-wt Fiberglass unless otherwise noted.

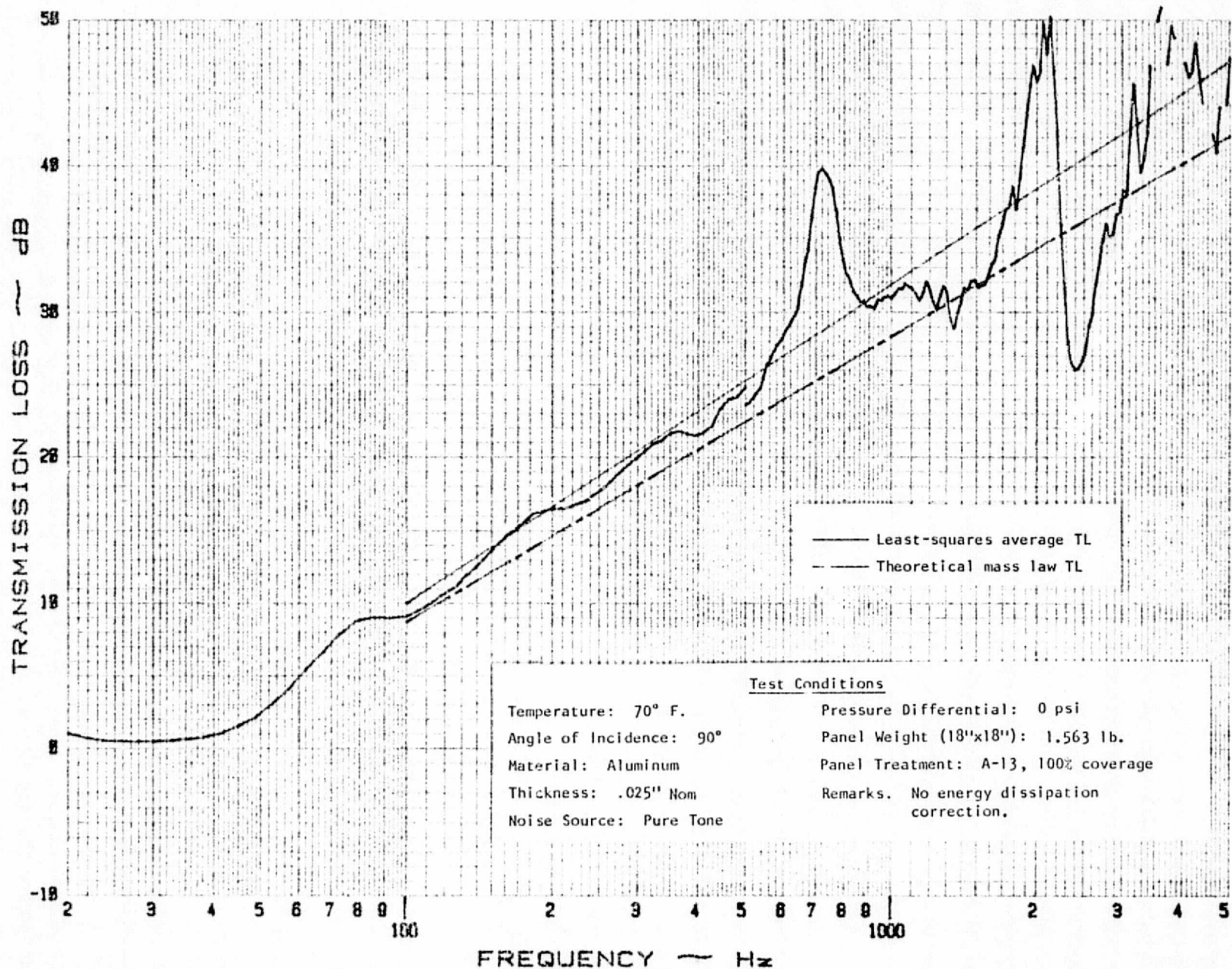
**Fiberglass Faces, Nomex Core. Faces are C-wt Fiberglass.

Table B.1 (continued)

<u>Figure</u>	<u>Material</u>	<u>Thickness (in.)</u>	<u>Test Condition or Treatment</u>
B.50	Aluminum	.040	3/32" x 17/32" slit
B.51	Aluminum	.040	3/32" x 1" slit
B.52	Aluminum	.040	5/32" x 1" slit
B.53	Aluminum	.040	reference solid panel
<u>Others</u>			
B.54	Polyurethane and matte fiberglass	5/16	bare (8), No DC
B.55	Polyurethane and matte fiberglass	5/16	bare (8), with DC
B.56	Steel	.020	L-136 woven asbestos substitute (9) with DC

Table B.2 List of Manufacturers of
Acoustic Materials

1. Specialty Composites Corp., Delaware Ind. Park,
Newark, Delaware.
2. Singer Partitions, Inc., 444 N. Lake Shore Dr.,
Chicago, Ill. 60611.
3. Insul-Coustic/Birma Corp., Jernee Mill Road,
Sayreville, N.J.
4. 3M Company, Minneapolis, Mn.
5. Foamade Industries, 1220 Morse Street,
Royal Oak, Michigan 48068.
6. Carney Insulations Corp., P.O. Box 1237,
Mankato, Mn. 56001.
7. Forty-Eight Insulations, Inc., Aurora, Ill.
8. Beech Aircraft Corp., Wichita, Ks. 67201.
(Honeycomb materials used in test panels were produced
by Hexcel Corporation, Dublin, CA 94566. Fiberglass
fabric used in some test panels was produced by U.S.
Polymeric, Inc., Santa Ana, California.)
9. Fiberfrax Co., address unknown. Material furnished
by Beech Aircraft Corp., Wichita, Ks. 67201.
10. Cessna Aircraft Company, Pawnee Division, Wichita,
Kansas (Panel Assembly).



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Figure B.1: Experimental Sound Transmission Loss of a Treated Aluminum Panel

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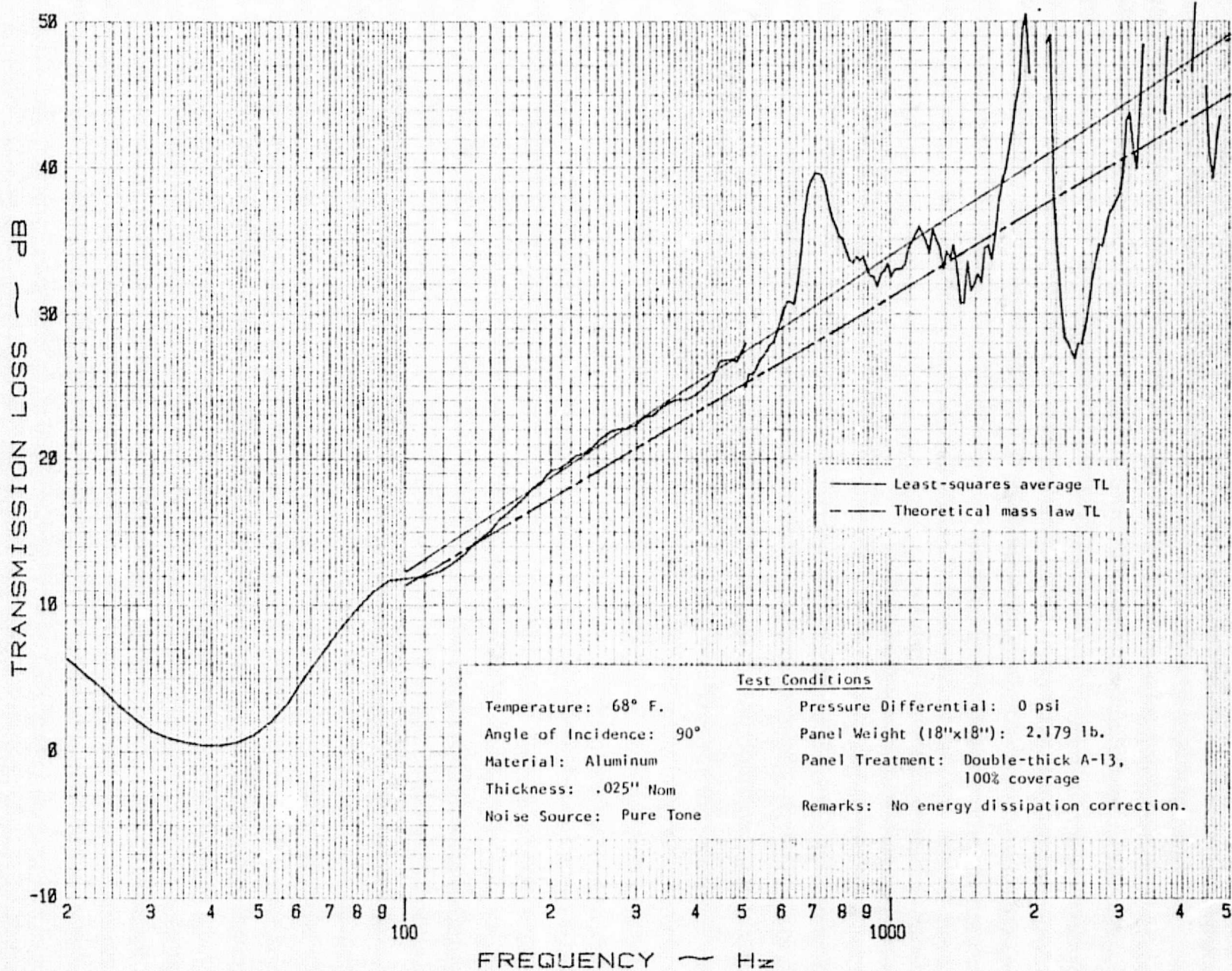


Figure B.2: Experimental Sound Transmission Loss of a Treated Aluminum Panel

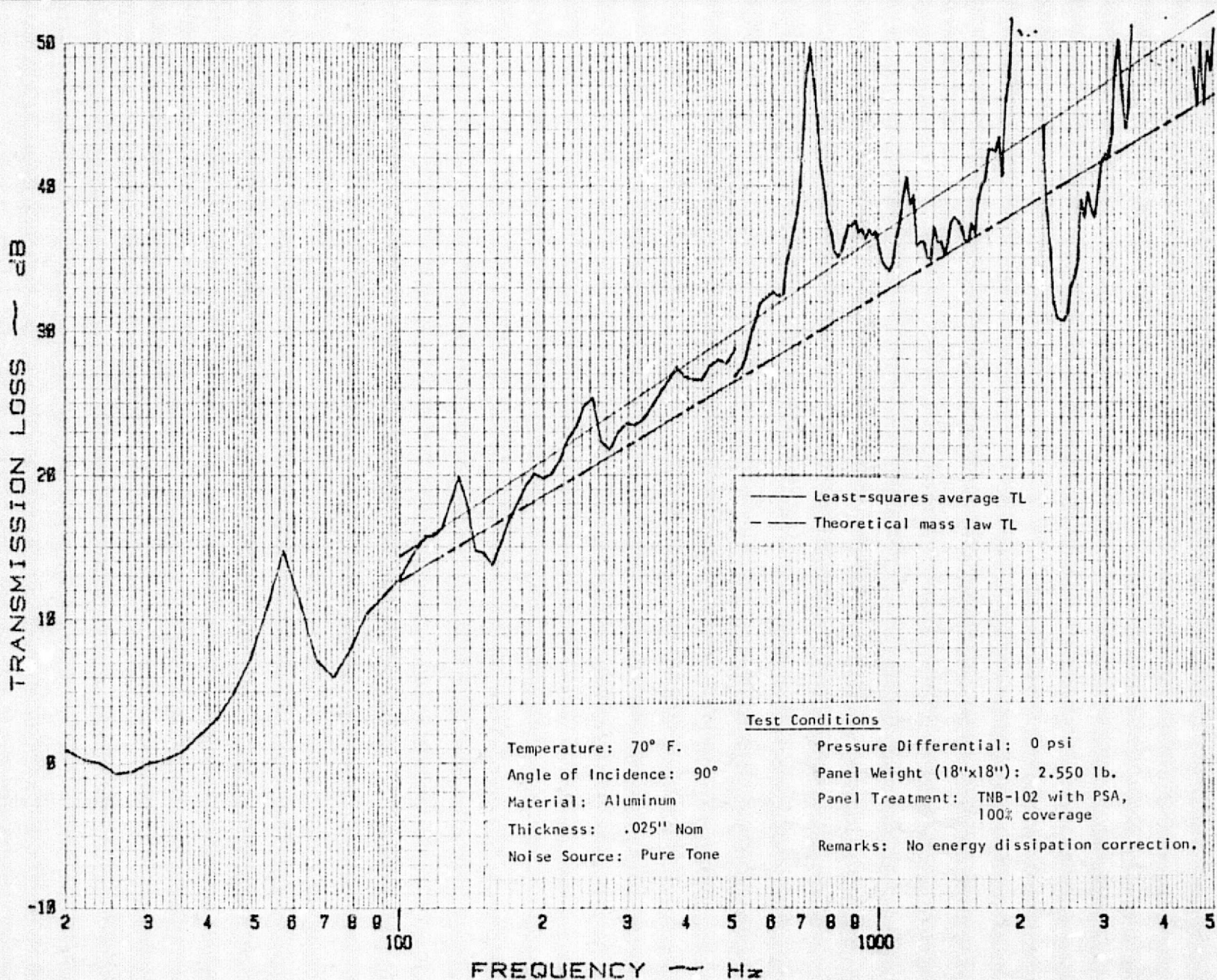


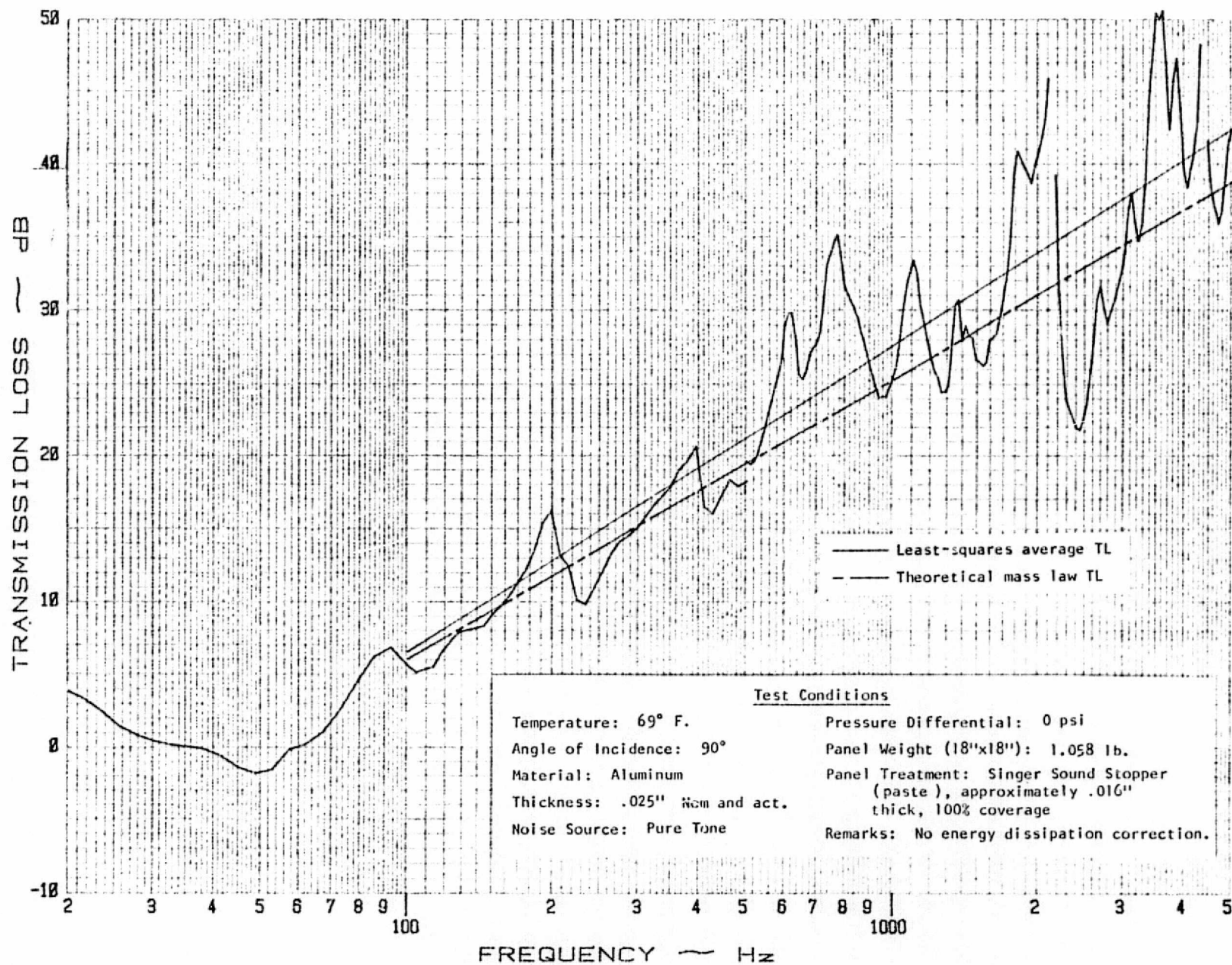
Figure B.3: Experimental Sound Transmission Loss of a Treated Aluminum Panel

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Figure B.4: Experimental Sound Transmission Loss of an Aluminum Panel

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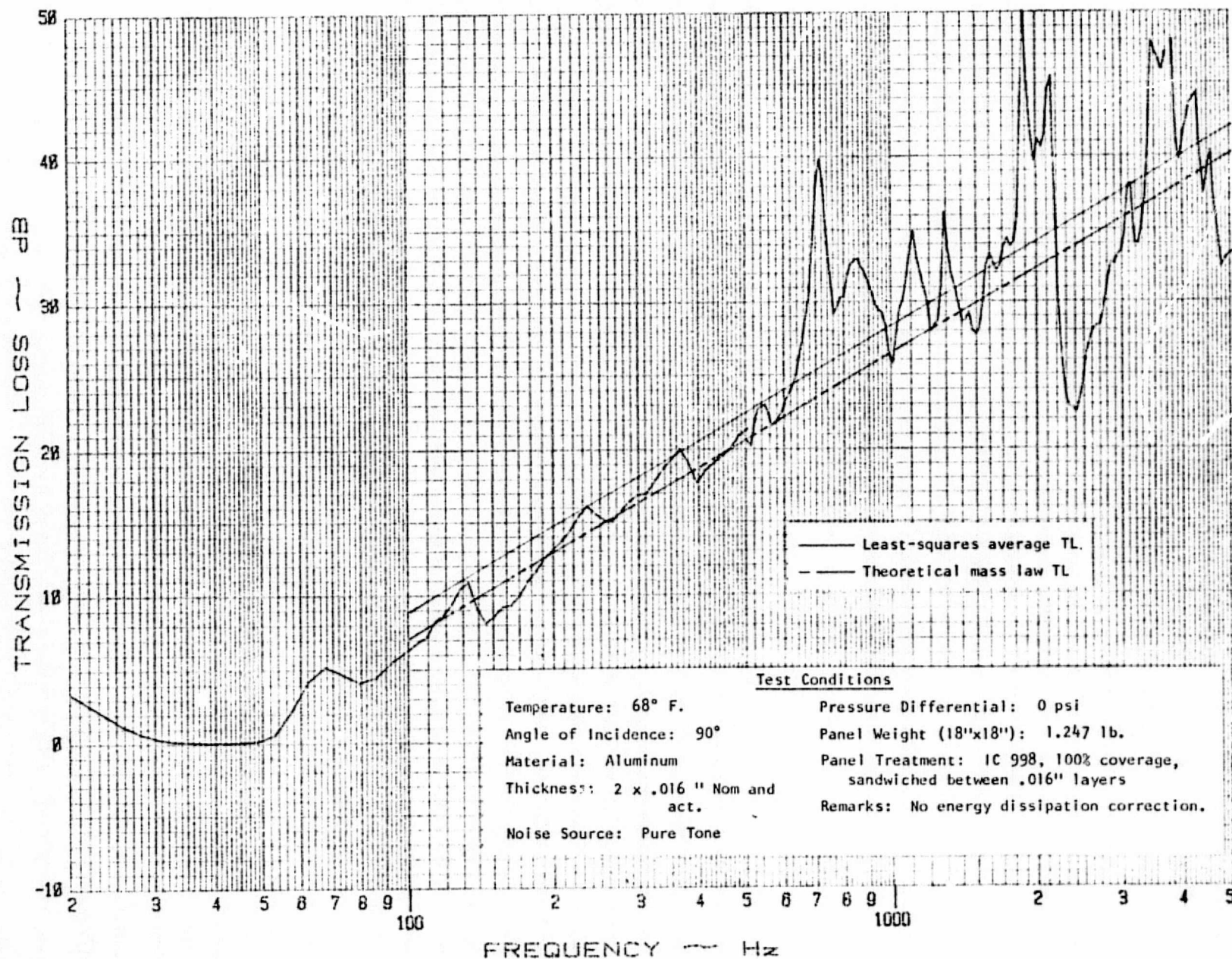


Figure B.5: Experimental Sound Transmission Loss of an Aluminum Sandwich Panel

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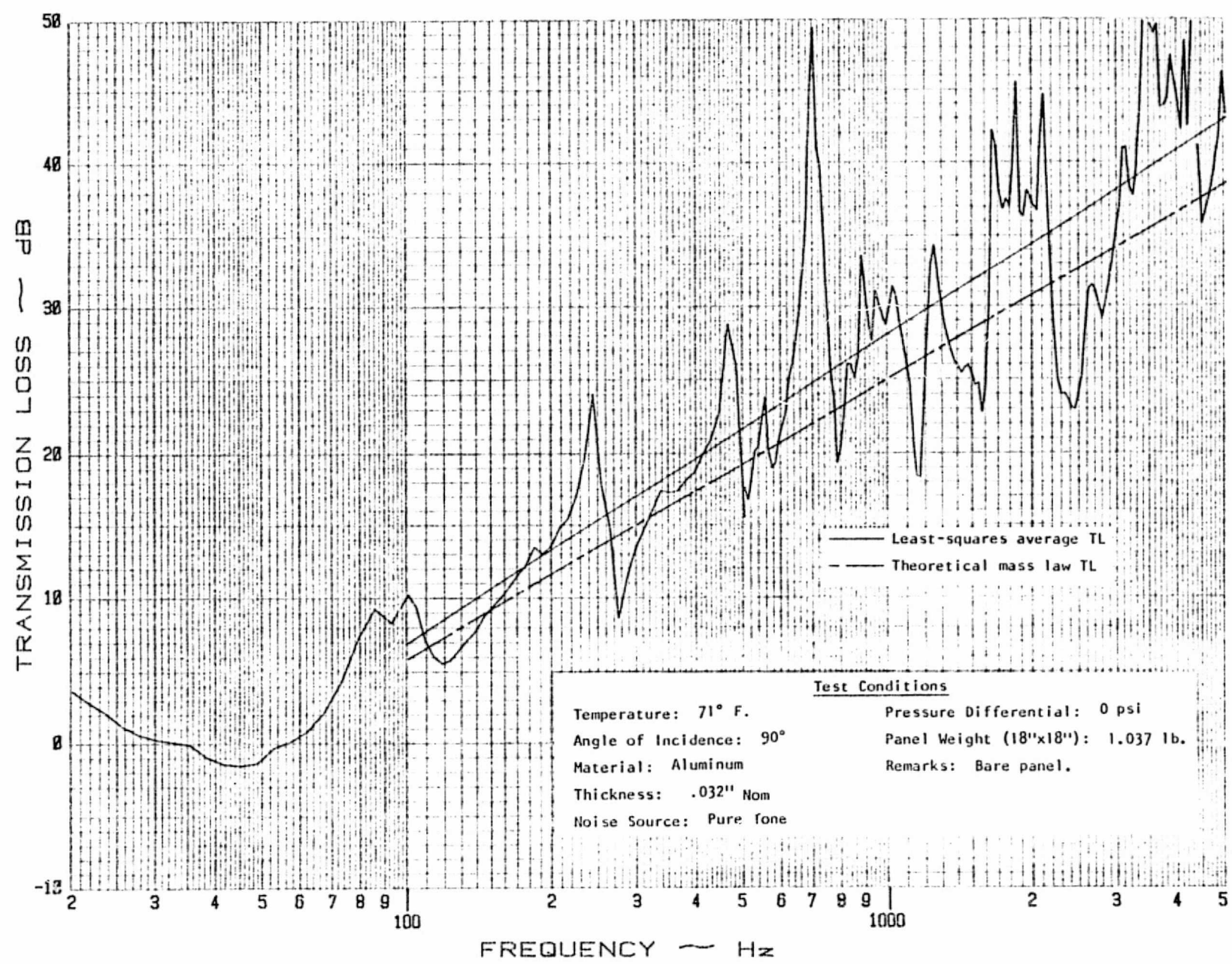


Figure B.6: Experimental Sound Transmission Loss of an Aluminum Panel

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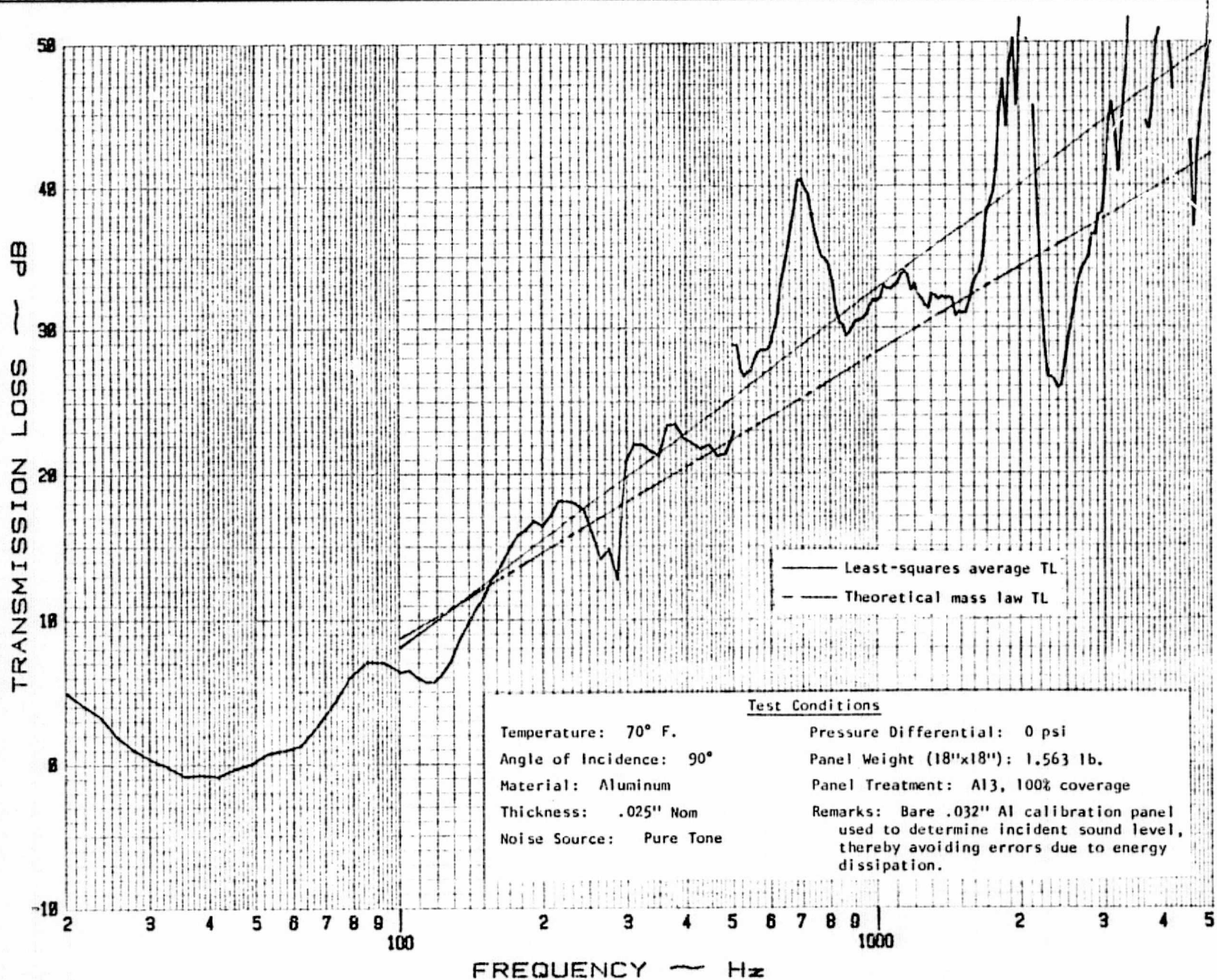


Figure B.7: Experimental Sound Transmission Loss of a Treated Aluminum Panel

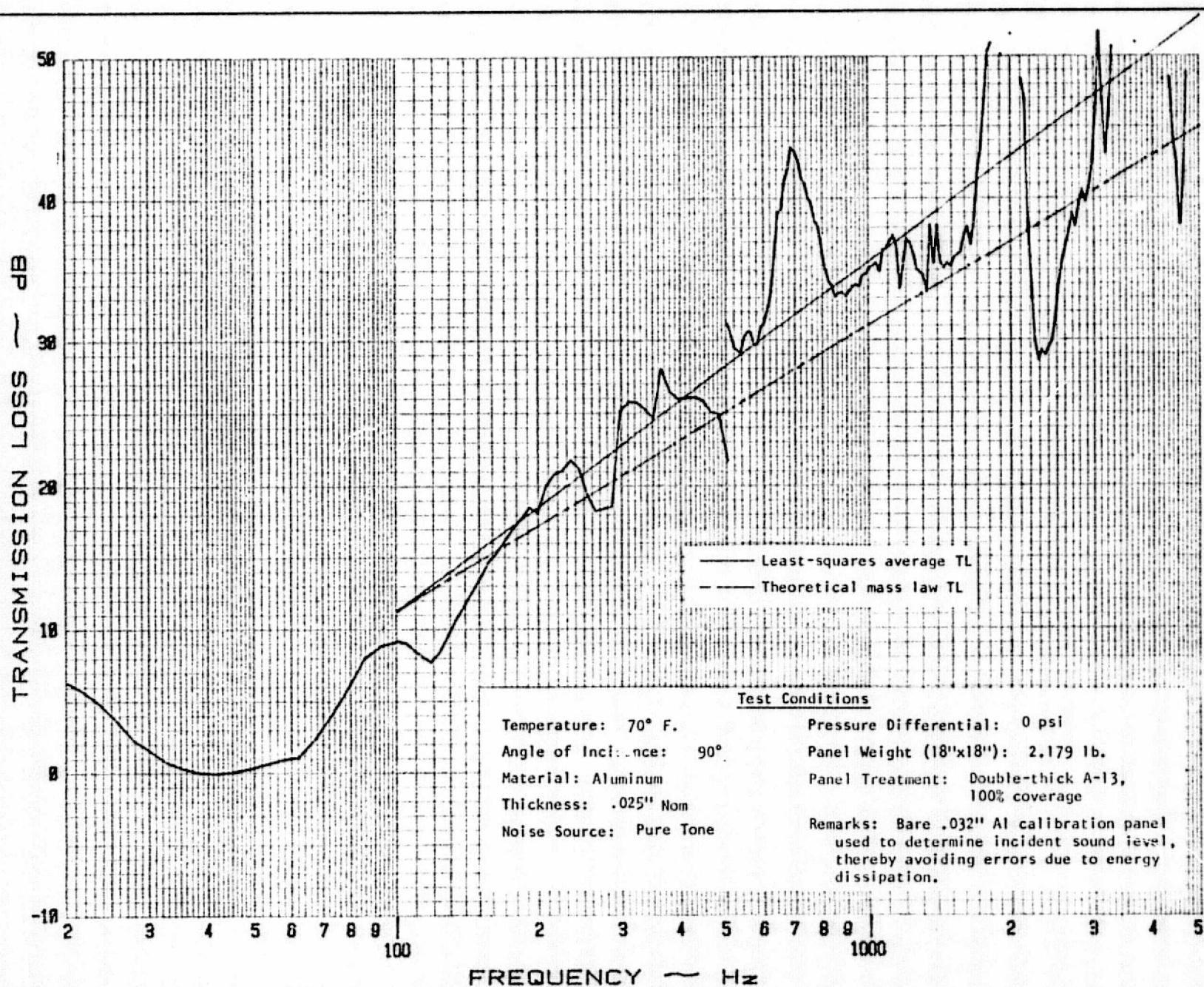


Figure B.8: Experimental Sound Transmission Loss of a Treated Aluminum Panel

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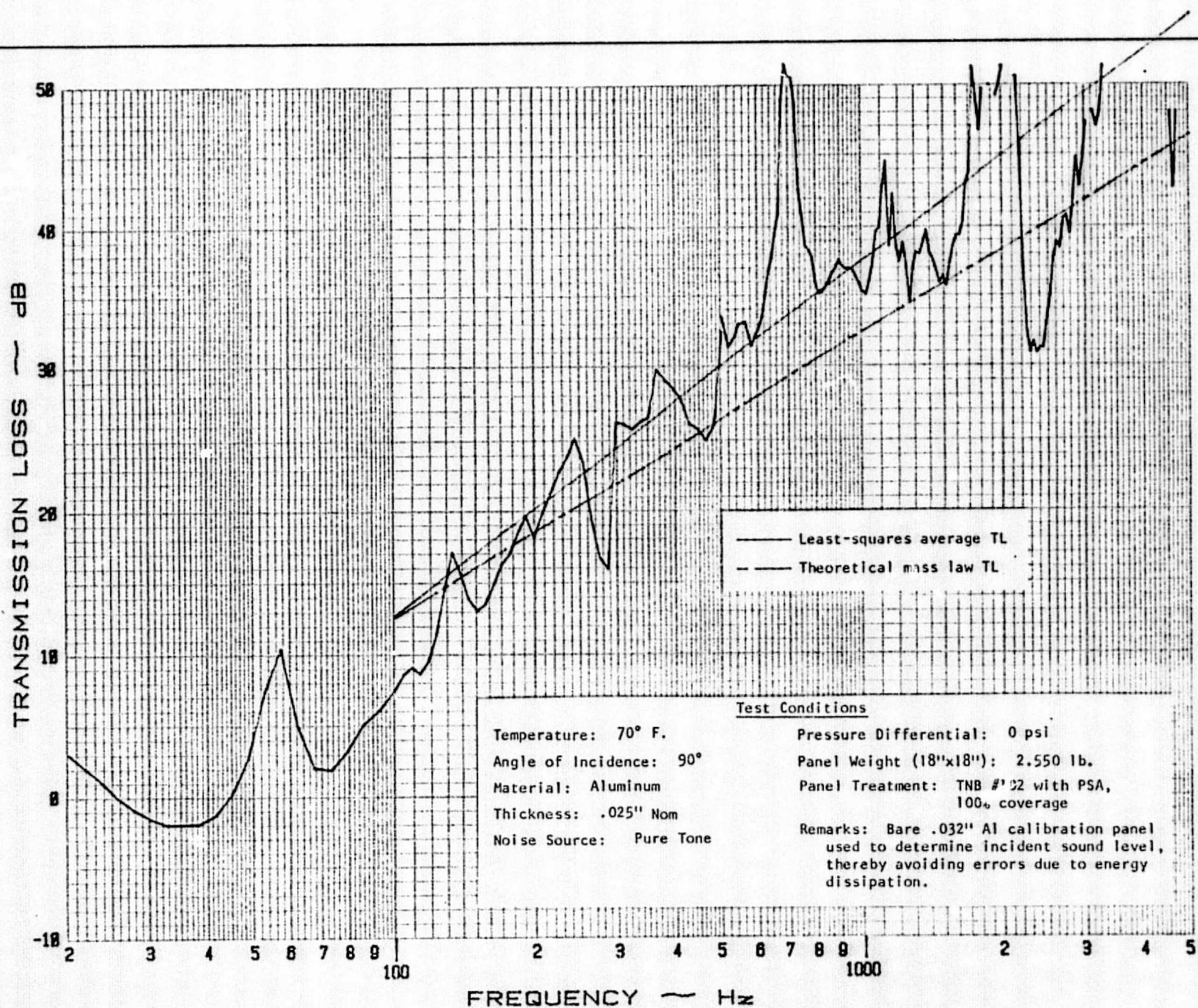


Figure B.9: Experimental Sound Transmission Loss of a Treated Aluminum Panel

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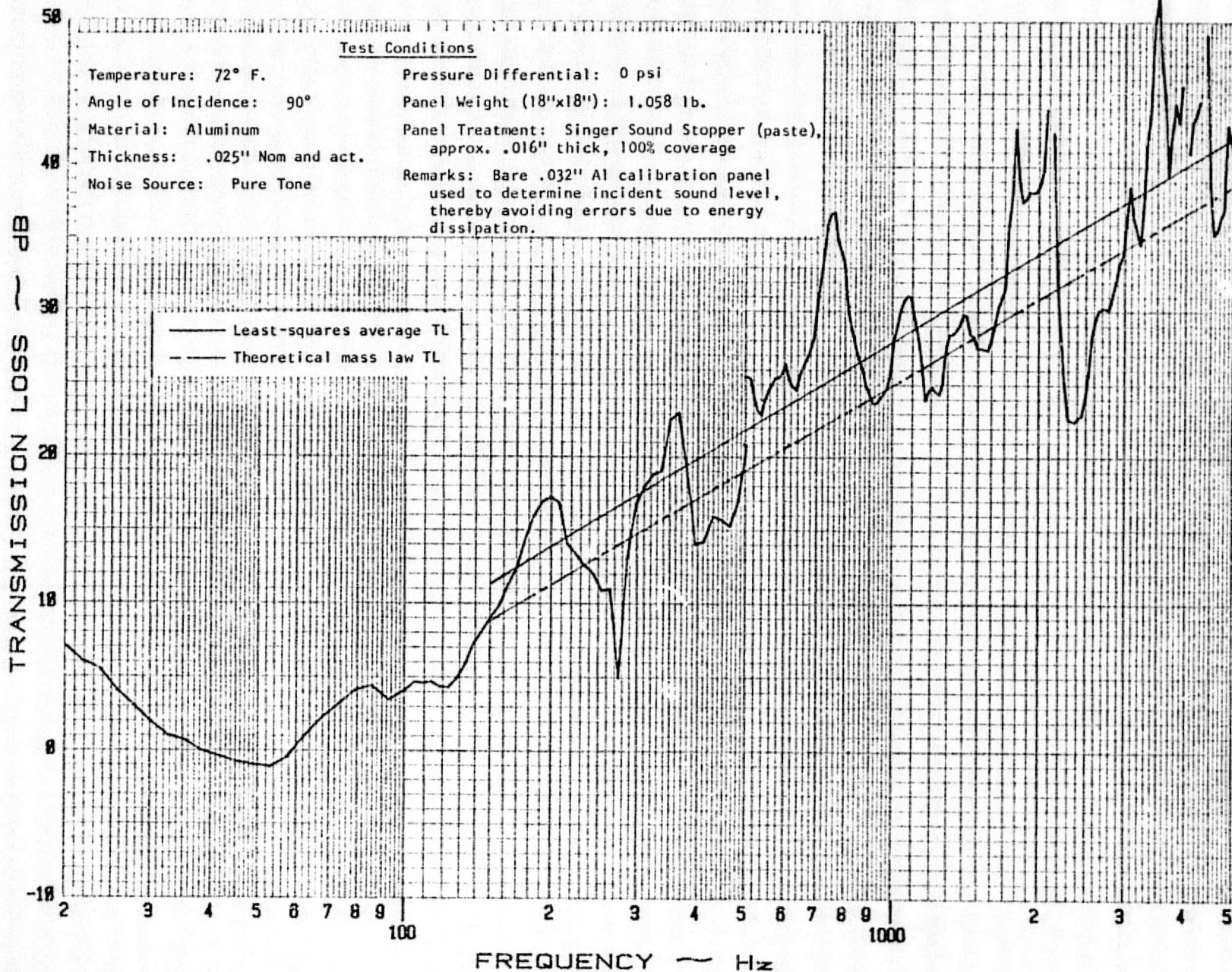


Figure B.10: Experimental Sound Transmission Loss of a Treated Aluminum Panel

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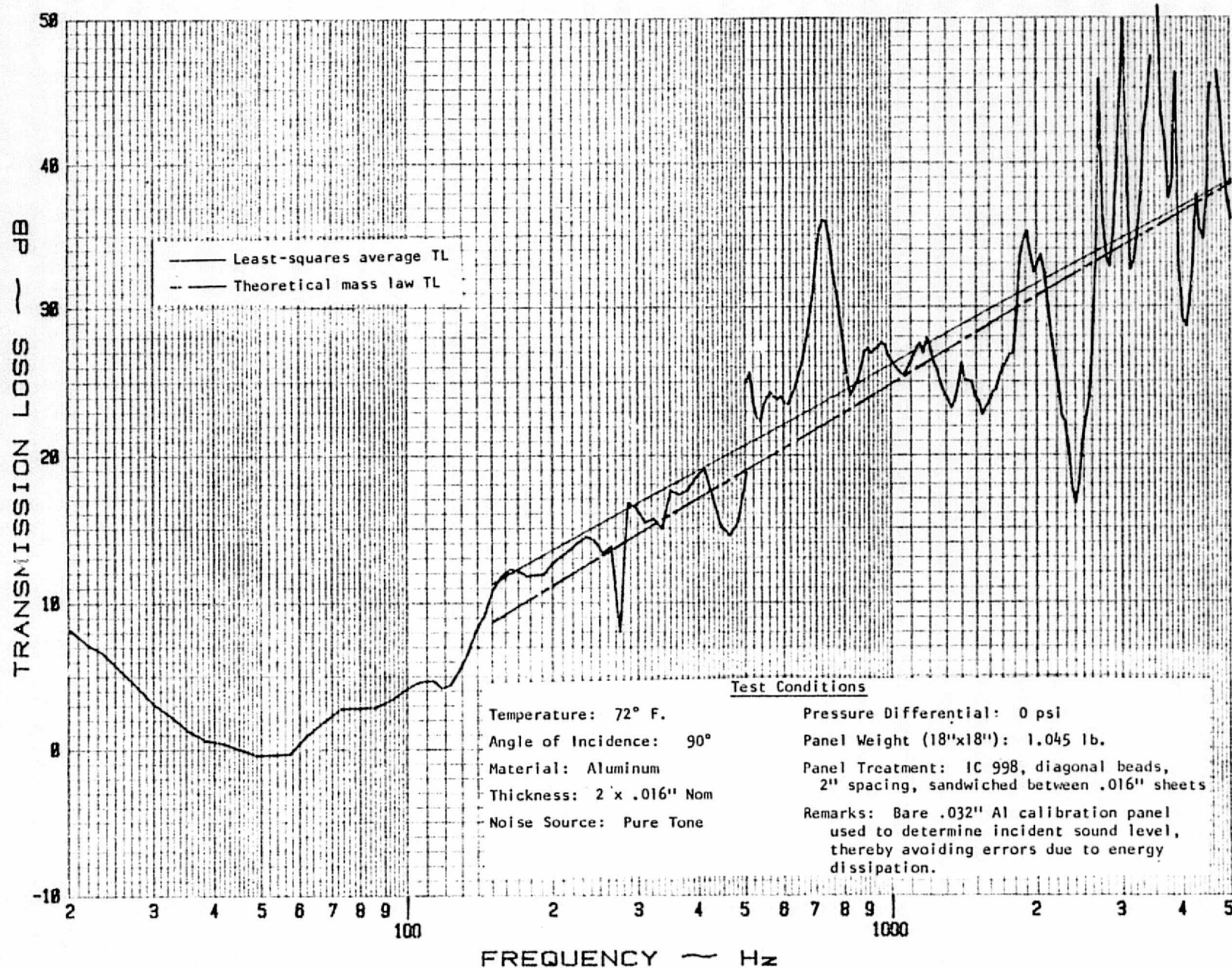
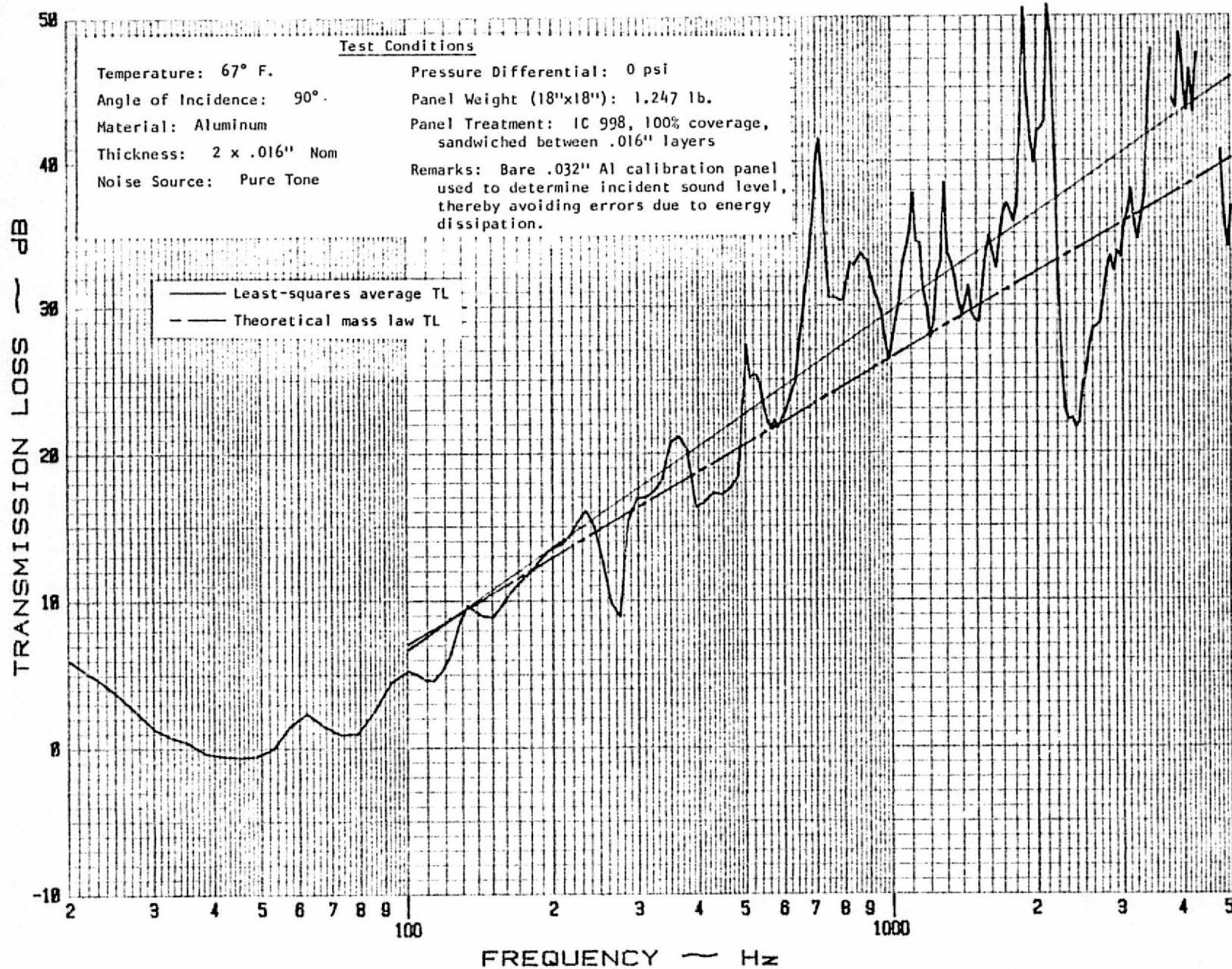


Figure B.11: Experimental Sound Trans-
mission Loss of an Aluminum Sandwich
Panel

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Figure B.12: Experimental Sound Transmission Loss of an Aluminum Sandwich Panel

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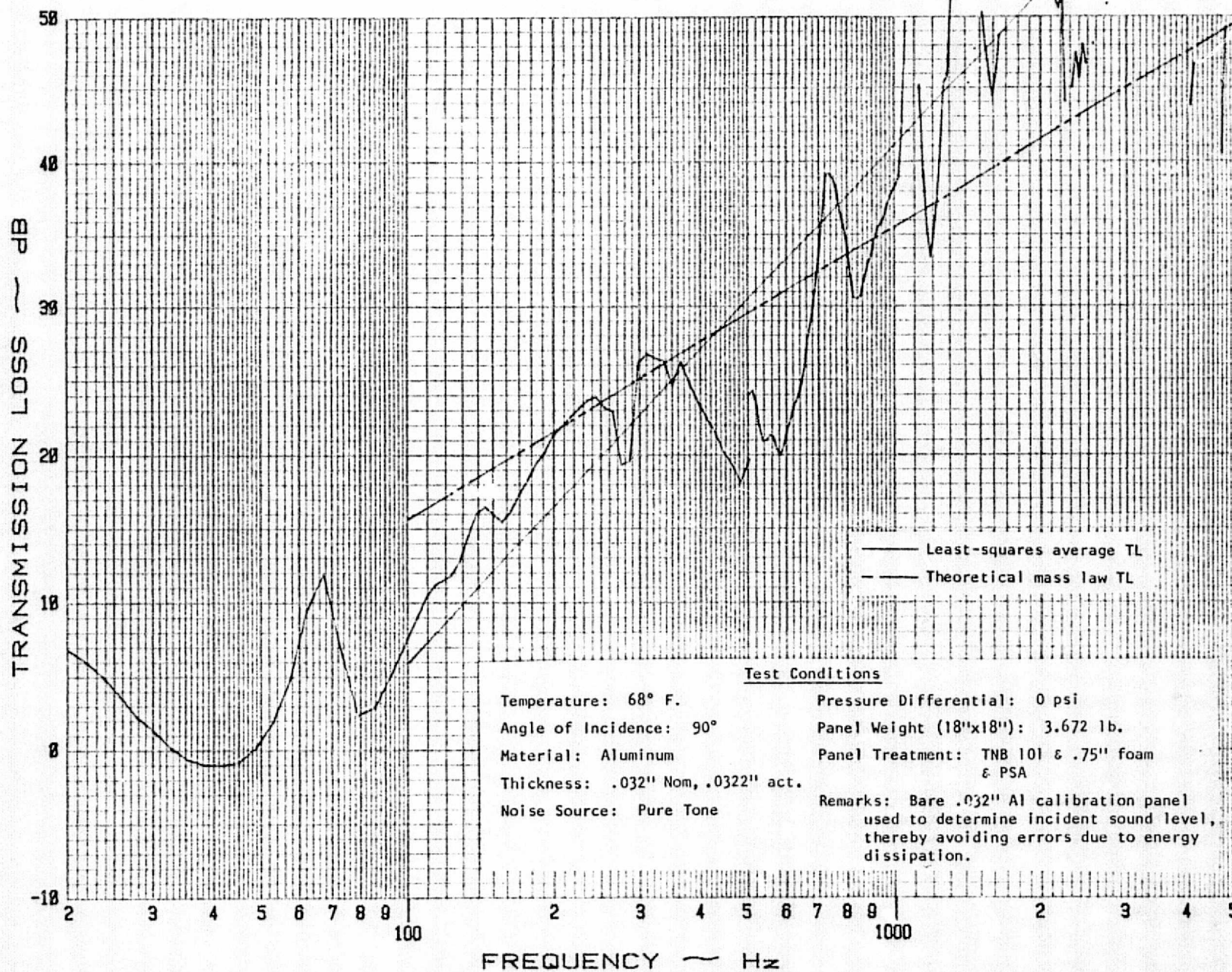


Figure B.13: Experimental Sound Transmission Loss of a Treated Aluminum Panel

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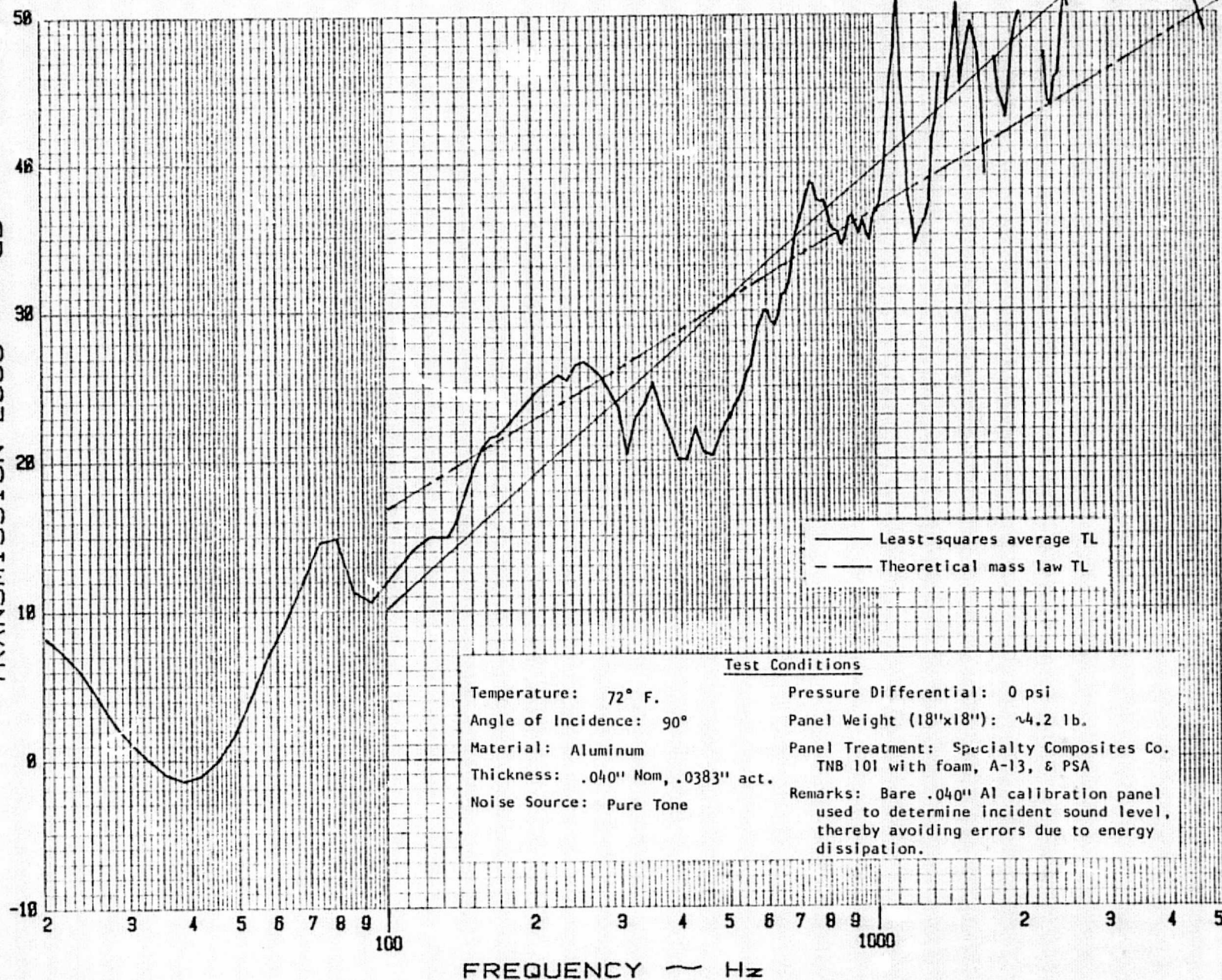
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BP ~ TRANSMISSION LOSS



Temperature: 72° F.
 Angle of Incidence: 90°
 Material: Aluminum
 Thickness: .040" Nom, .0383" act.
 Noise Source: Pure Tone

Test Conditions

Pressure Differential: 0 psi
 Panel Weight (18"x18"): ~4.2 lb.
 Panel Treatment: Specialty Composites Co.
 TNB 101 with foam, A-13, & PSA
 Remarks: Bare .040" Al calibration panel
 used to determine incident sound level,
 thereby avoiding errors due to energy
 dissipation.

Figure B.14: Experimental Sound Trans-

mission Loss of a Treated Aluminum Panel

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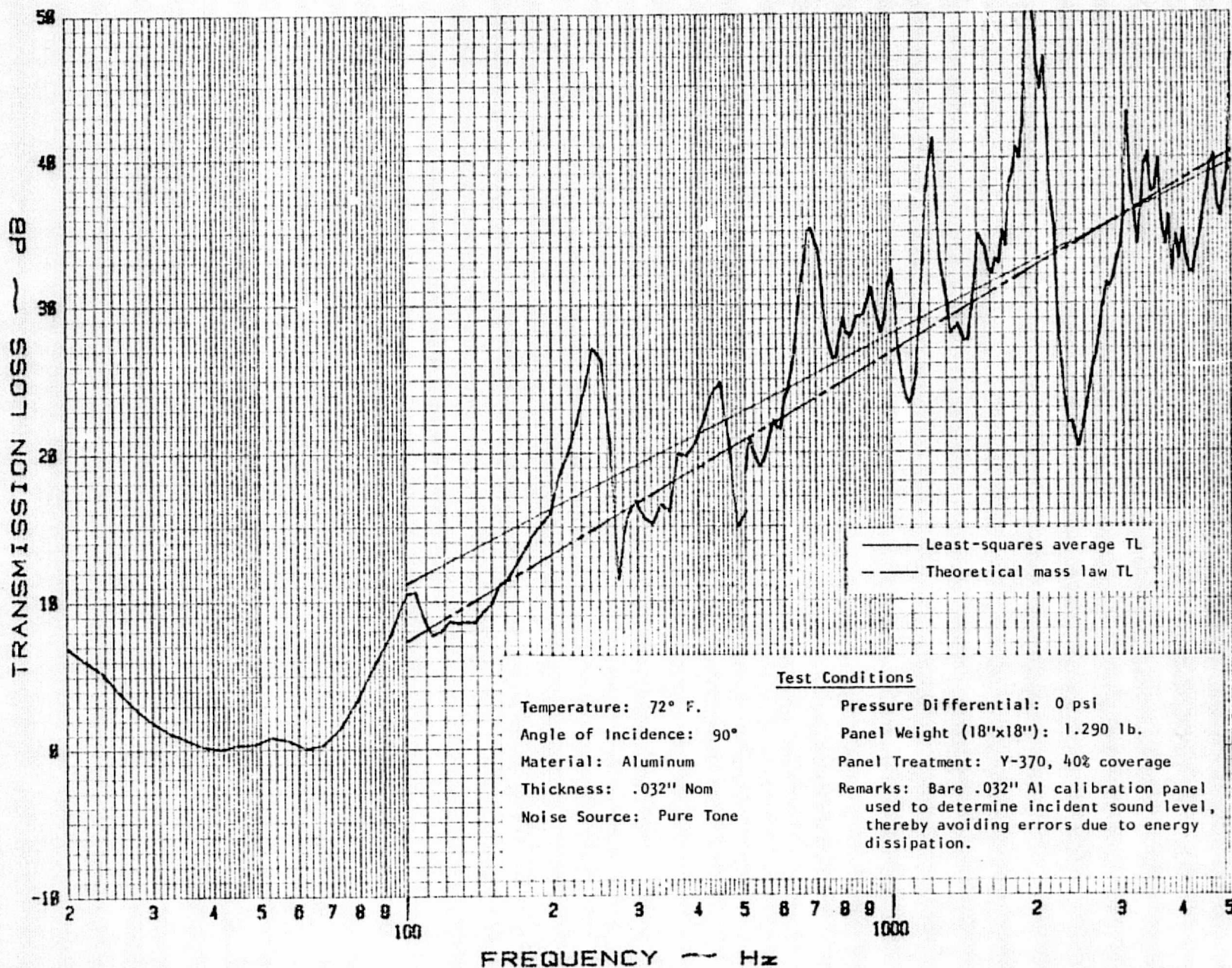


Figure B.15: Experimental Sound Transmission Loss of a Treated Aluminum Panel

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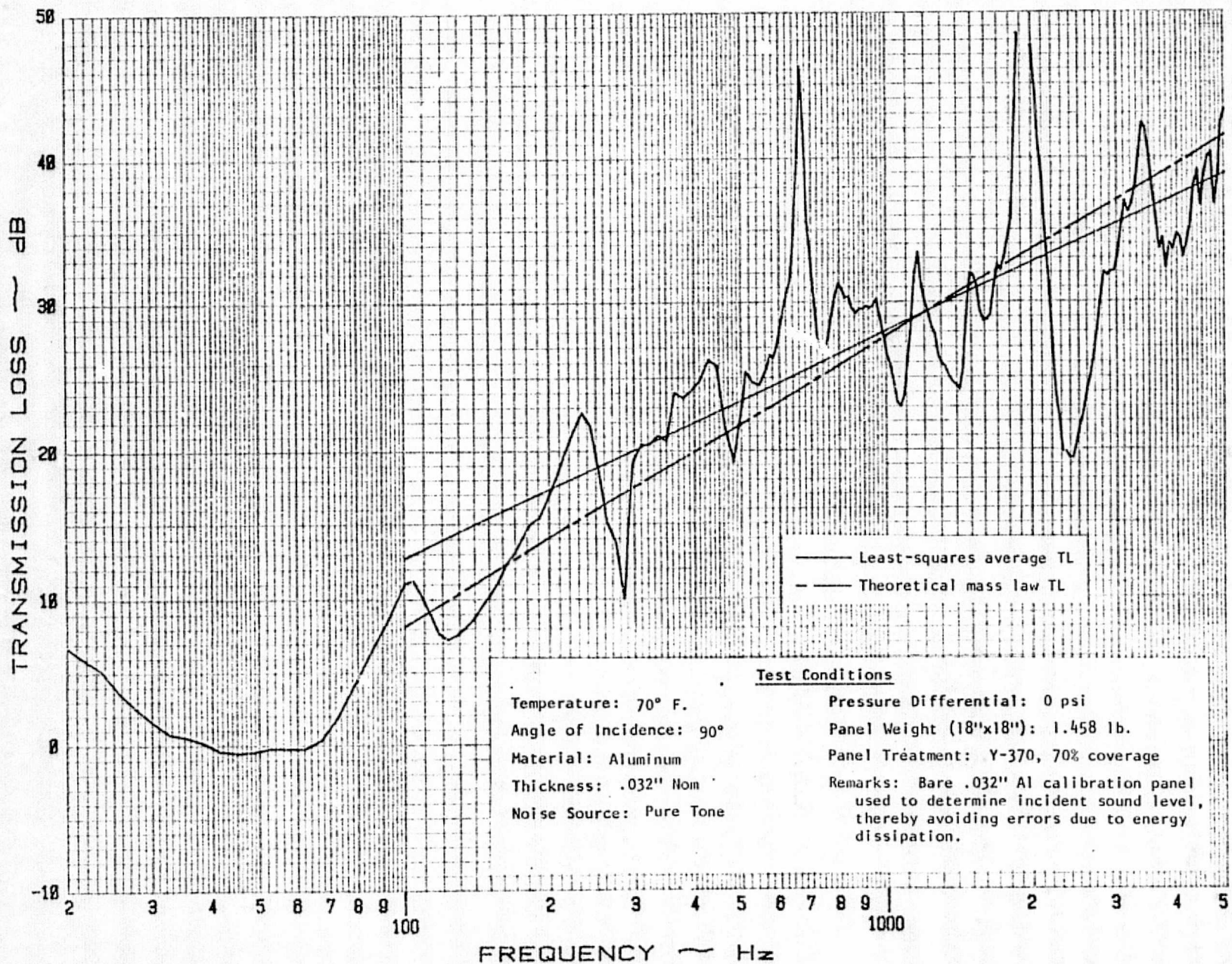


Figure B.16: Experimental Sound Transmission Loss of a Treated Aluminum Panel

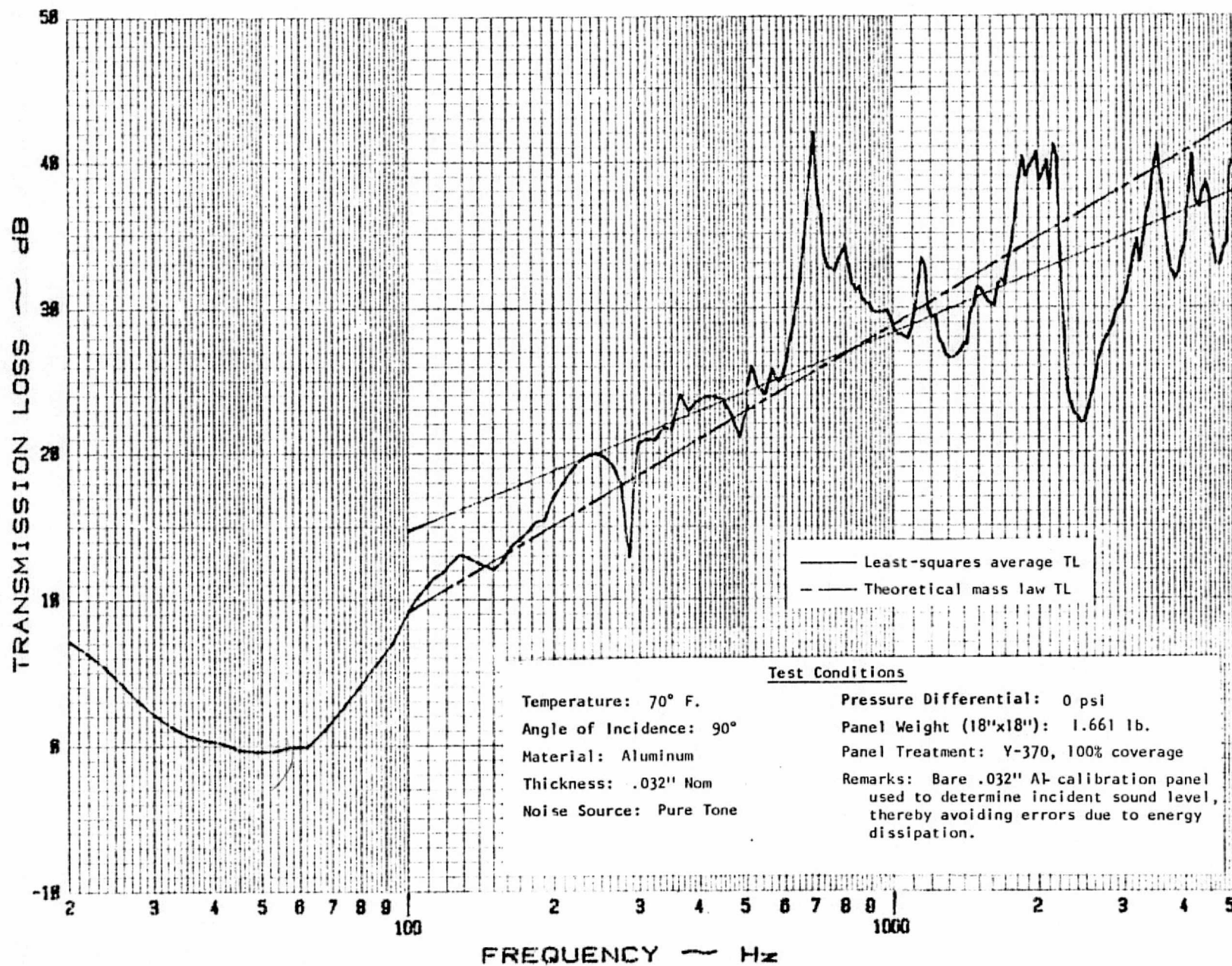
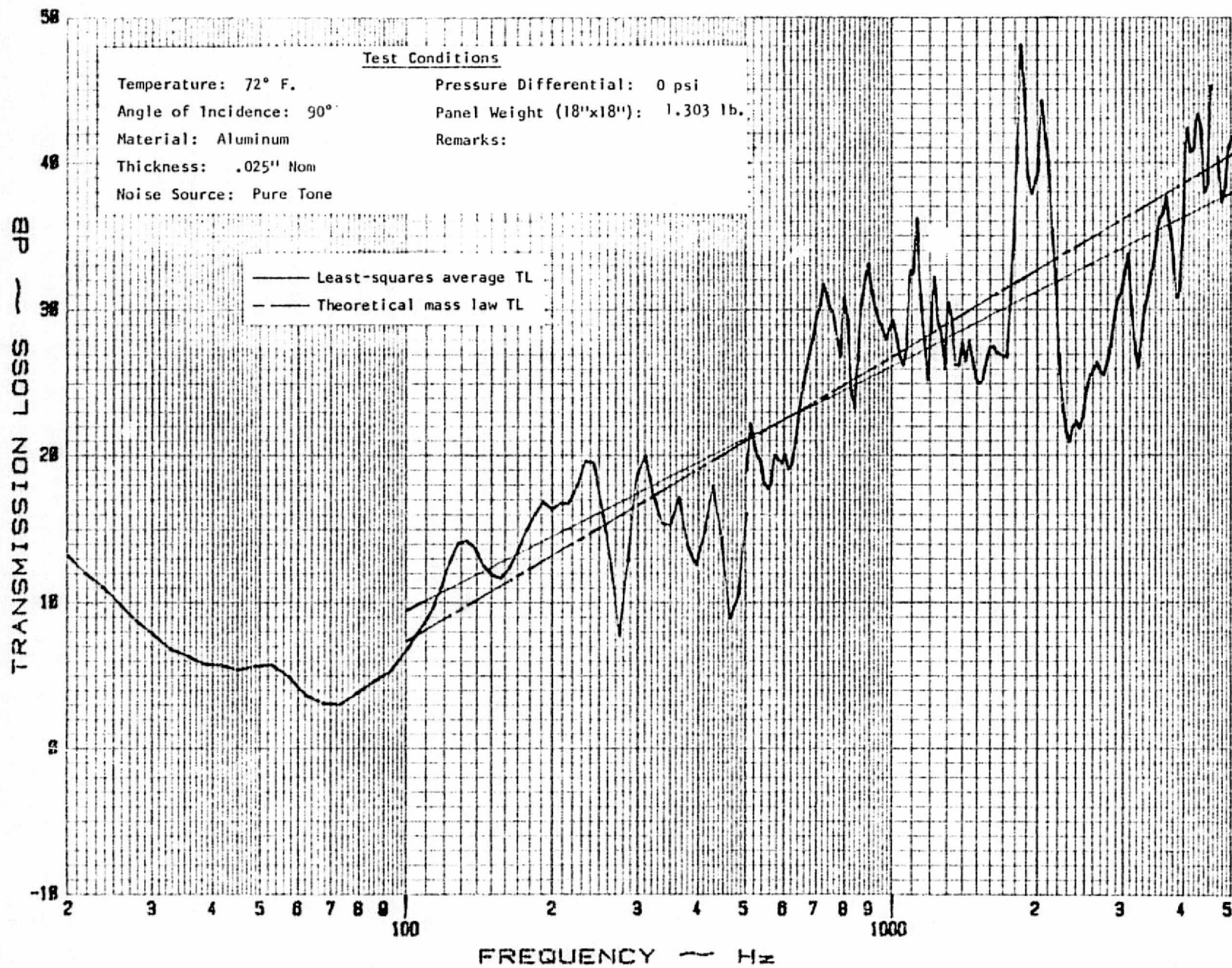


Figure B.17: Experimental Sound Transmission Loss of a Treated Aluminum Panel

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Figure B.18: Experimental Sound Transmission Loss of a Stiffened Aluminum Panel

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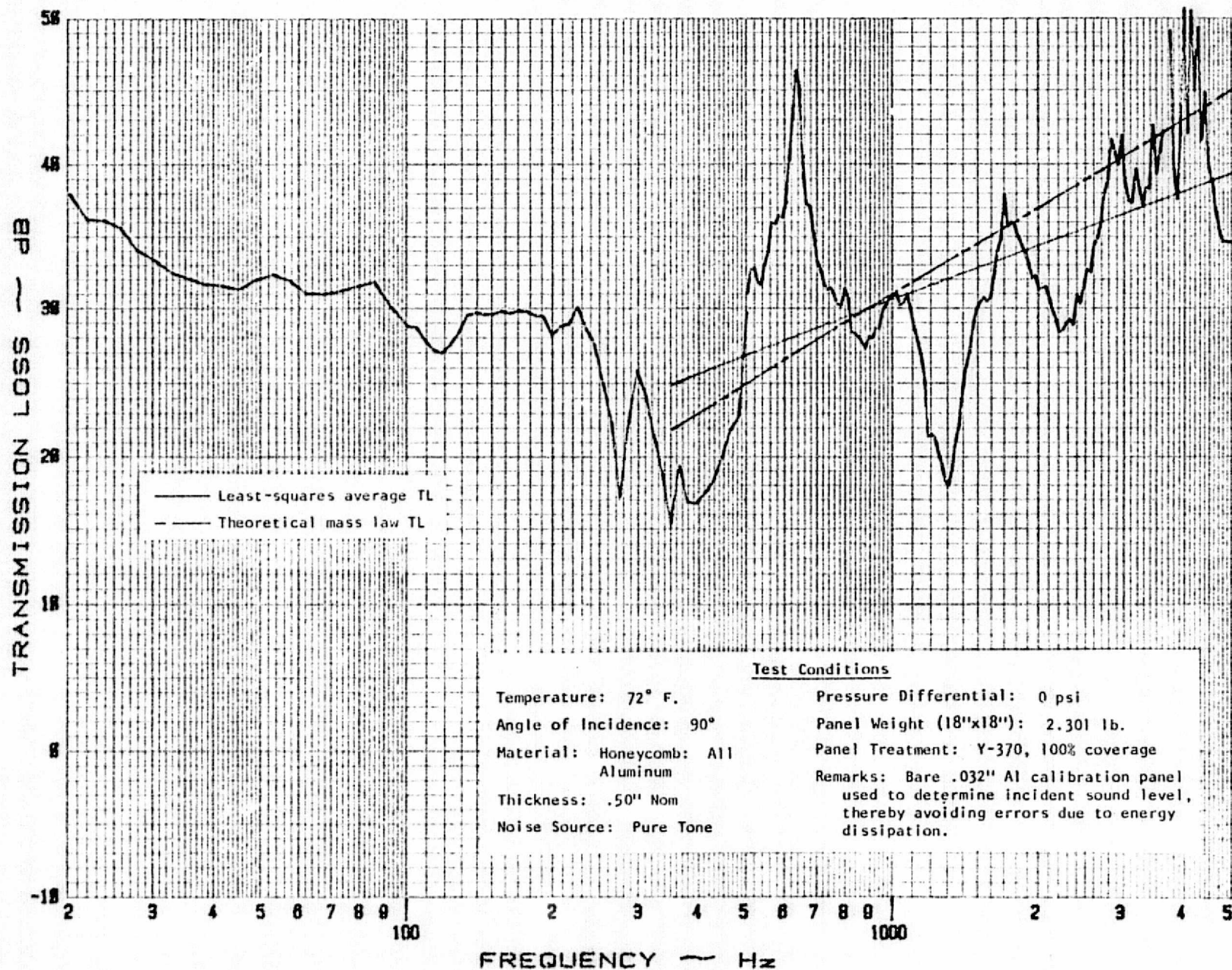


Figure B.19: Experimental Sound Transmission Loss of a Treated Honeycomb Panel

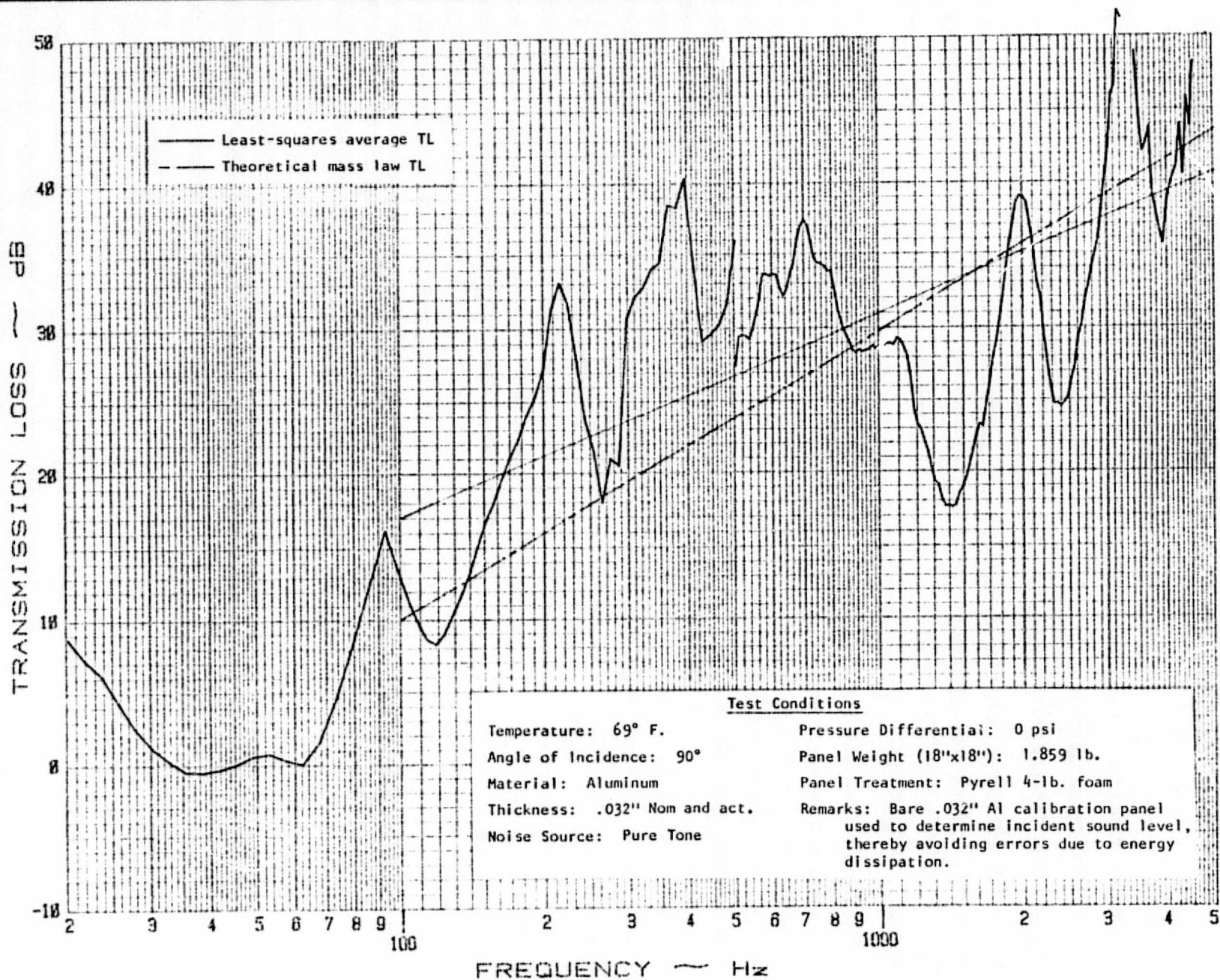
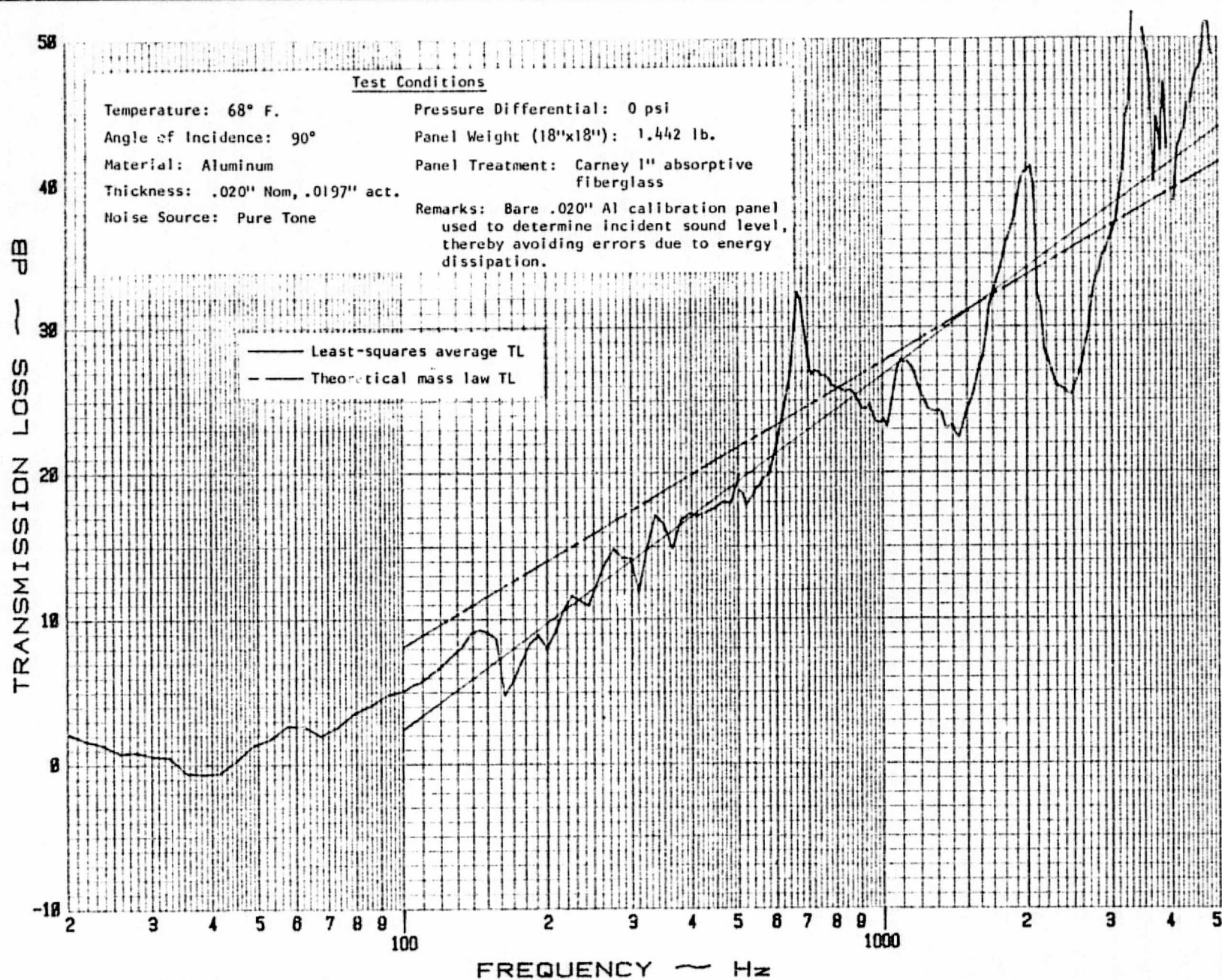


Figure B.20: Experimental Sound Transmission Loss of a Treated Aluminum Panel

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Test Conditions

Temperature: 72° F. Pressure Differential: 0 psi
Angle of Incidence: 90° Panel Weight (18"x18"): 2.350 lb.
Material: Aluminum Panel Treatment: Forty-Eight Insulations Co.
Thickness: .020" Nom, .0190" act. Fiberglass, 6 lb/ft³, 100% coverage
Noise Source: Pure Tone Remarks: Bare .020" Al calibration panel
used to determine incident sound level,
thereby avoiding errors due to energy
dissipation.

The graph plots Transmission Loss (TL) in dB on the y-axis (ranging from -10 to 50) against Frequency in Hz on a logarithmic x-axis (ranging from 2 to 5000). Two data series are shown: a solid line for 'Least-squares average TL' and a dashed line for 'Theoretical mass law TL'. The solid line shows significant fluctuations, particularly between 100 and 1000 Hz, with peaks and valleys. The dashed line is a smooth, straight line with a positive slope, representing the theoretical mass law. The solid line generally follows the trend of the dashed line but with increasing deviation at higher frequencies.

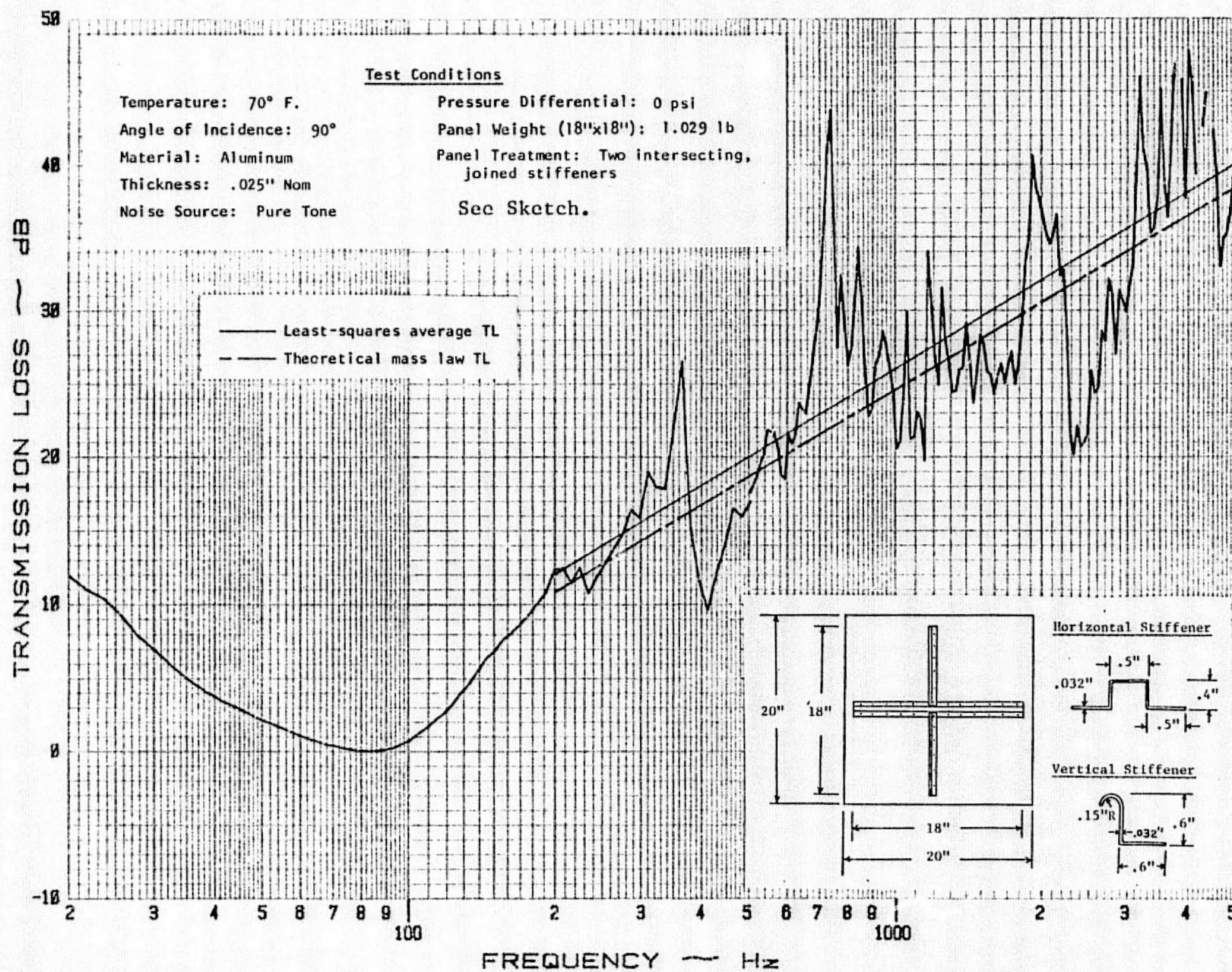
Frequency (Hz)	Least-squares average TL (dB)	Theoretical mass law TL (dB)
2	5	5
10	10	10
100	15	15
200	18	18
500	25	25
1000	30	30
2000	35	35
5000	45	45

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Figure B.24: Experimental Sound Transmission Loss of a Stiffened Aluminum Panel				PAGE 120
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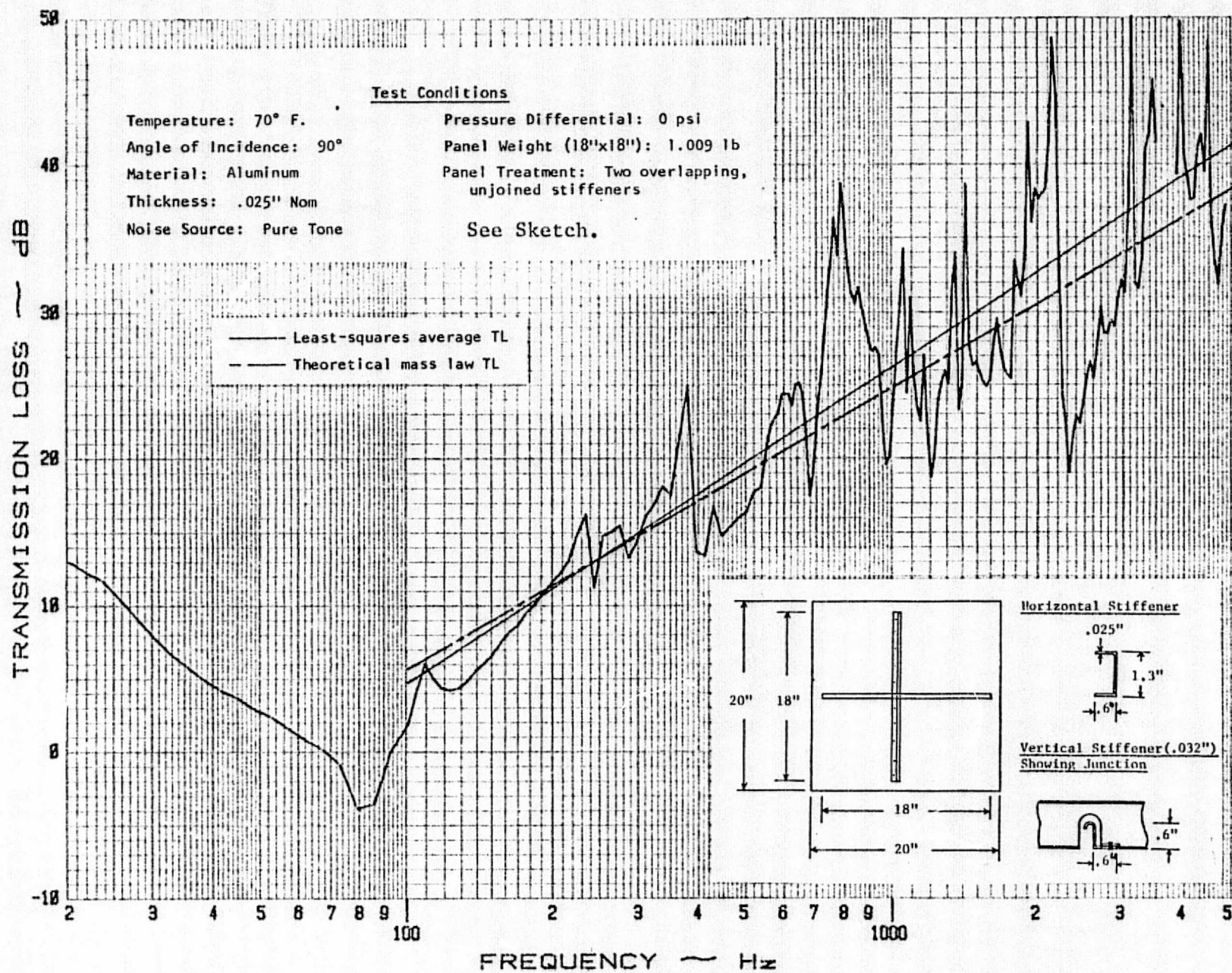


Figure B.25: Experimental Sound Transmission Loss of a Stiffened Aluminum Panel

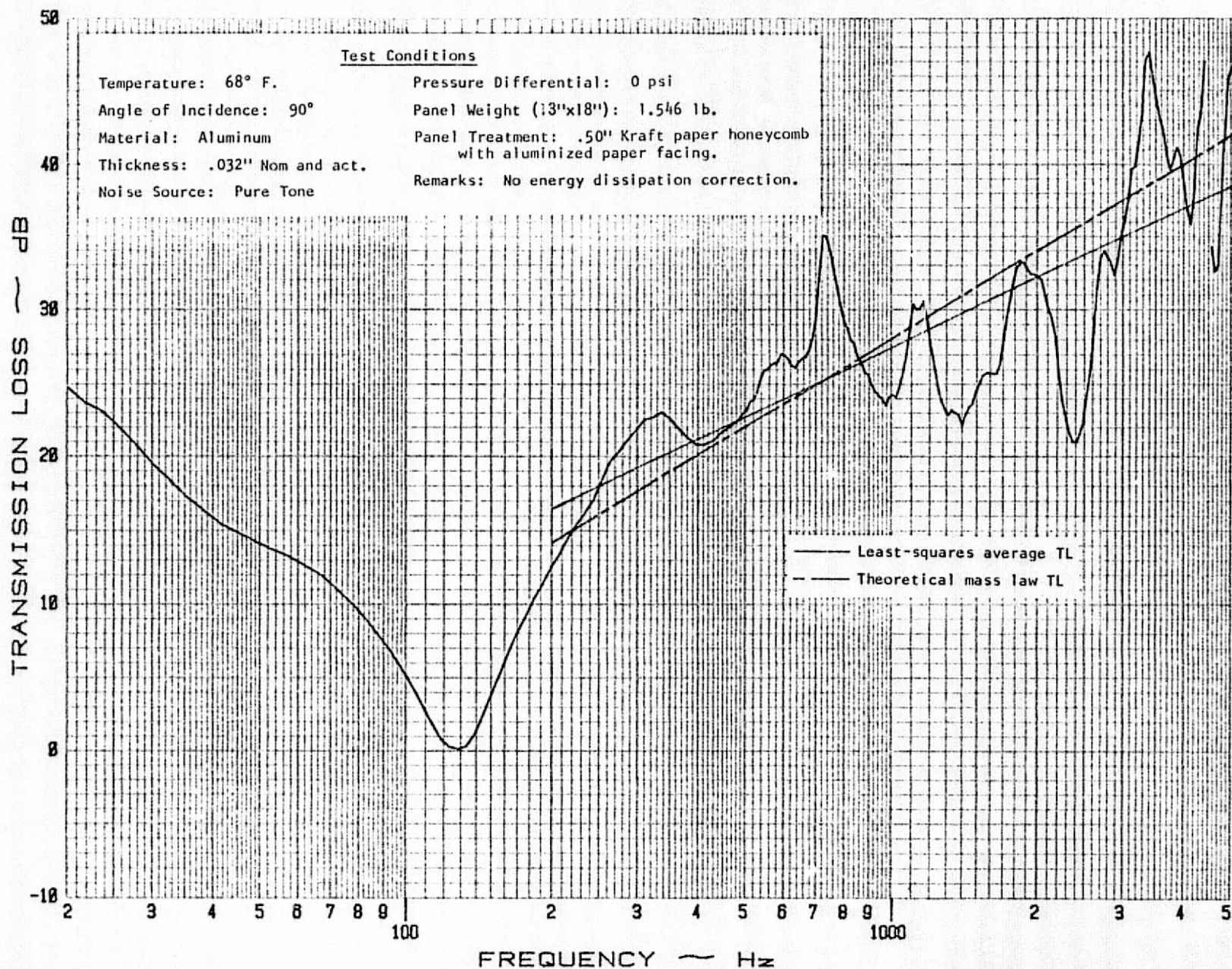


Figure B.26: Experimental Sound Transmission Loss of a Honeycomb Panel

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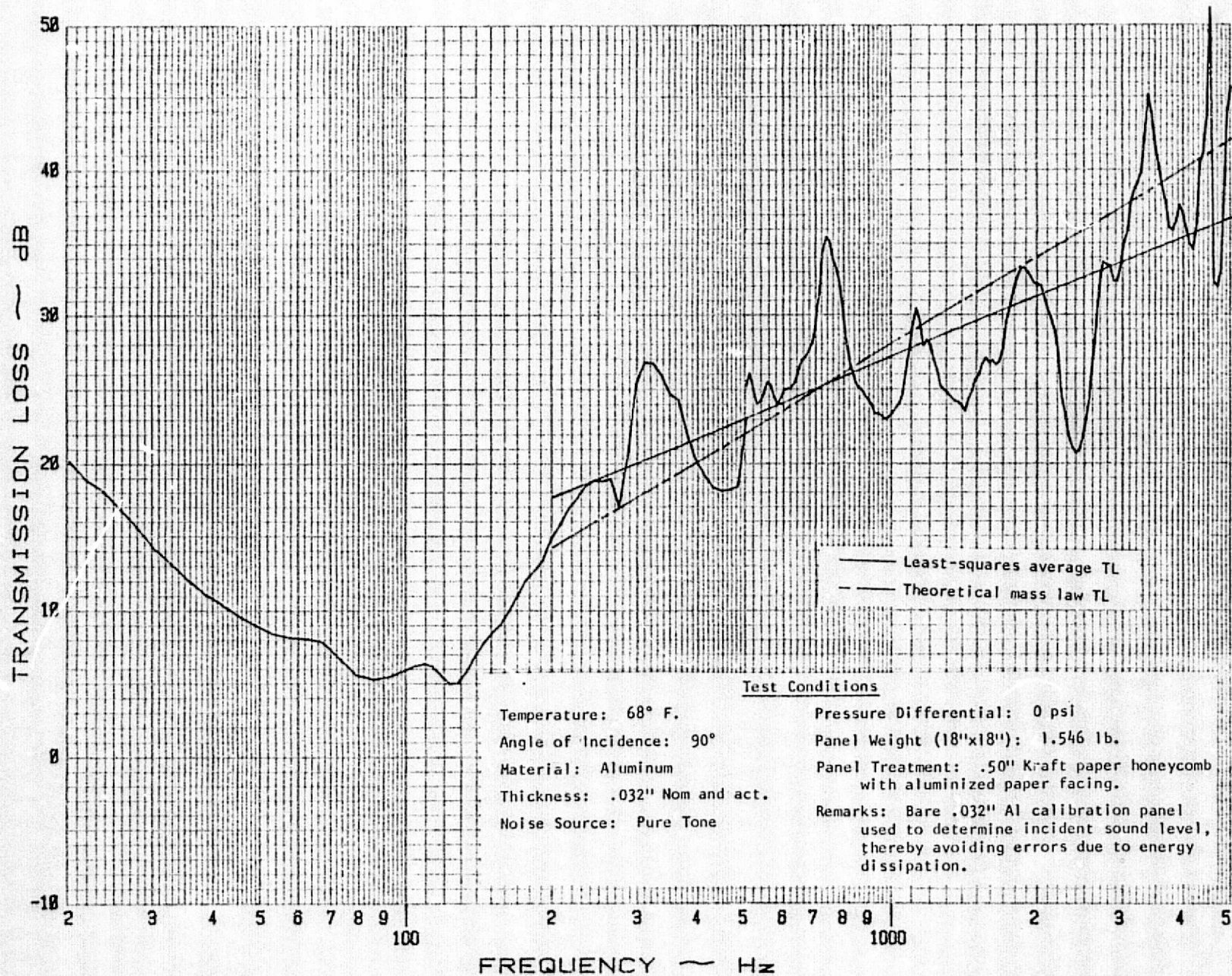


Figure B.27: Experimental Sound Transmission Loss of a Honeycomb Panel

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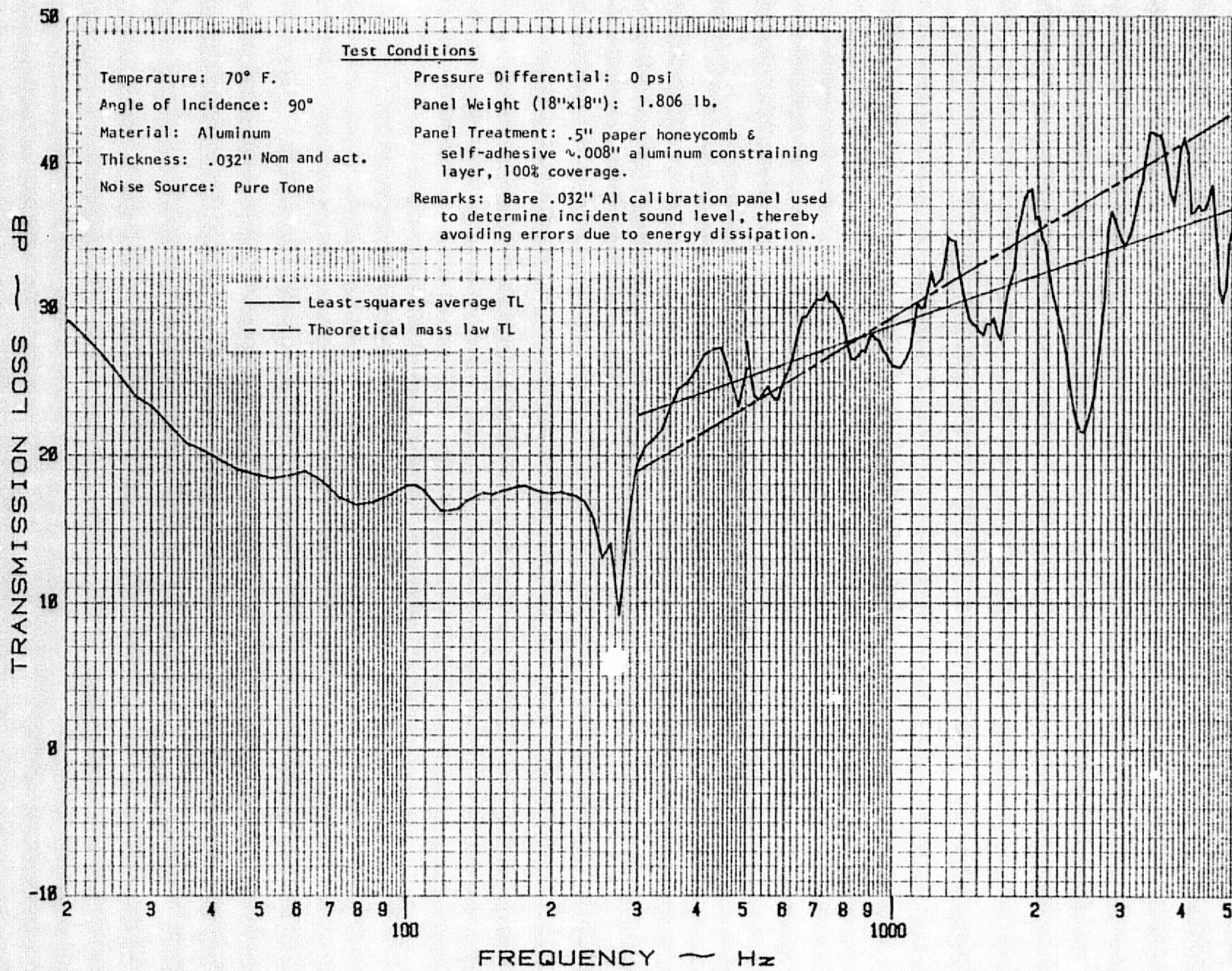


Figure B.28: Experimental Sound Transmission Loss of a Honeycomb Panel

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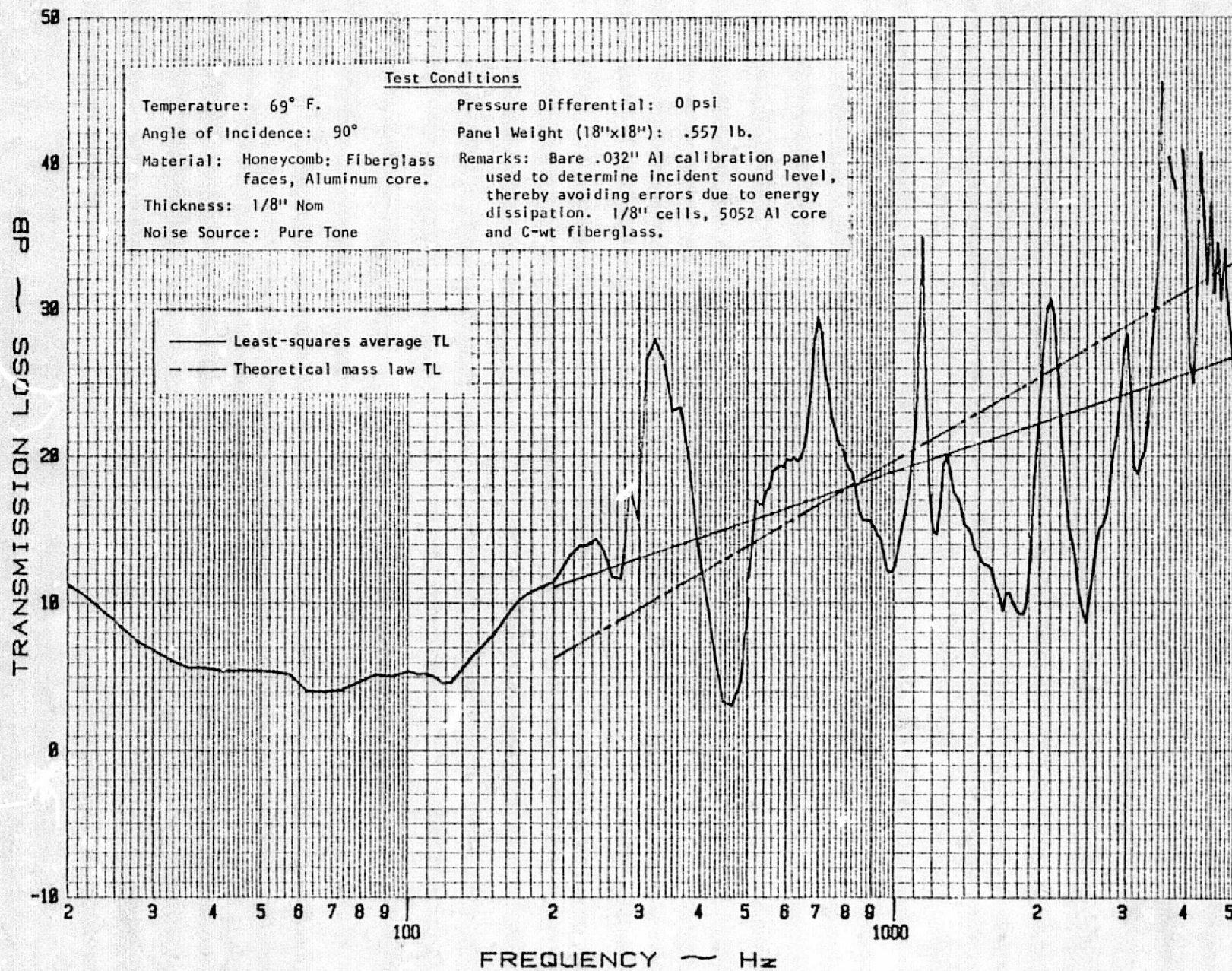


Figure B.29: Experimental Sound Transmission Loss of a Honeycomb Panel

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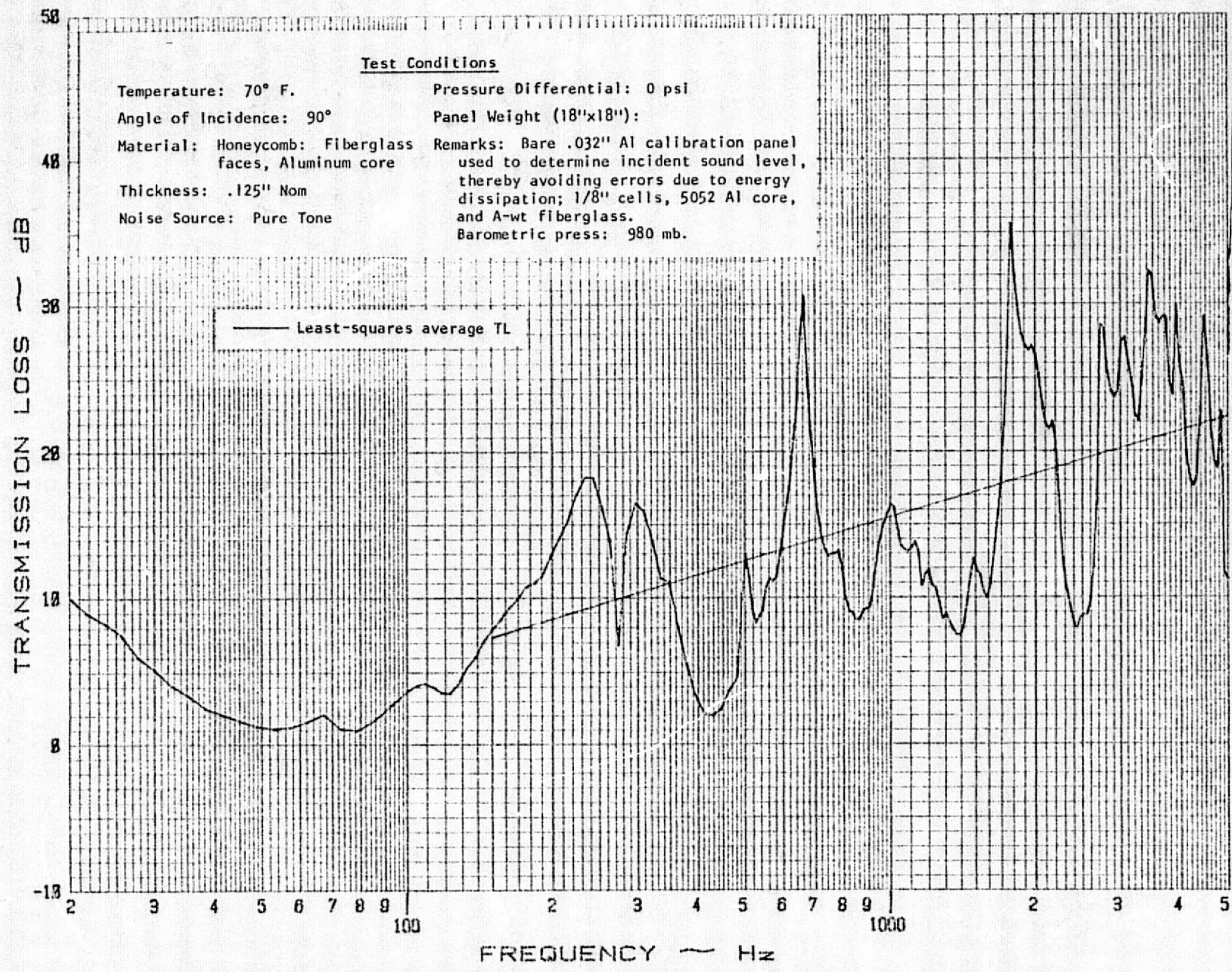
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CALC		REVISED	DATE	Figure B.30: Experimental Sound Transmission Loss of a Honeycomb Panel	UNIVERSITY OF KANSAS	PAGE B.35
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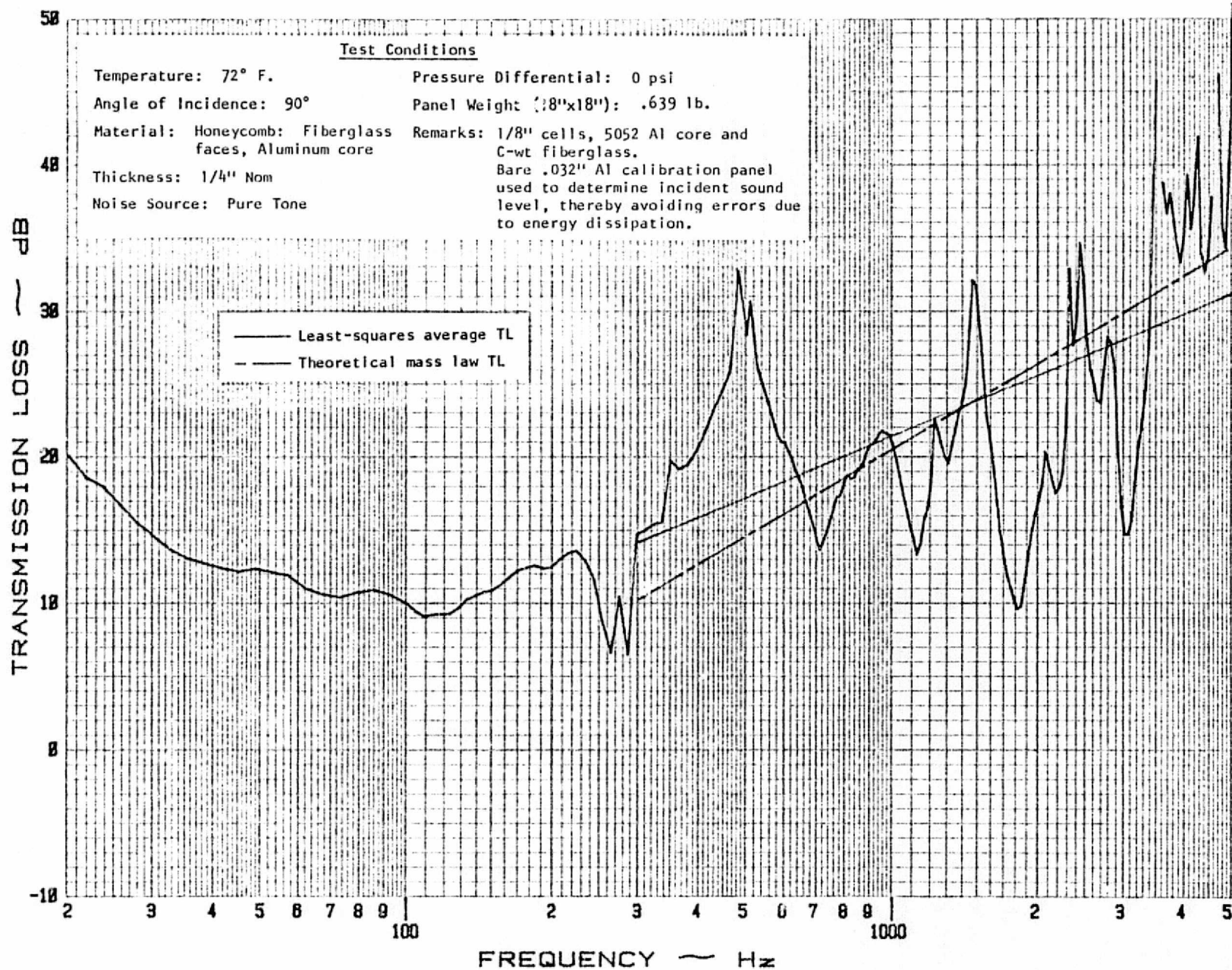


Figure B.31: Experimental Sound Transmission Loss of a Honeycomb Panel

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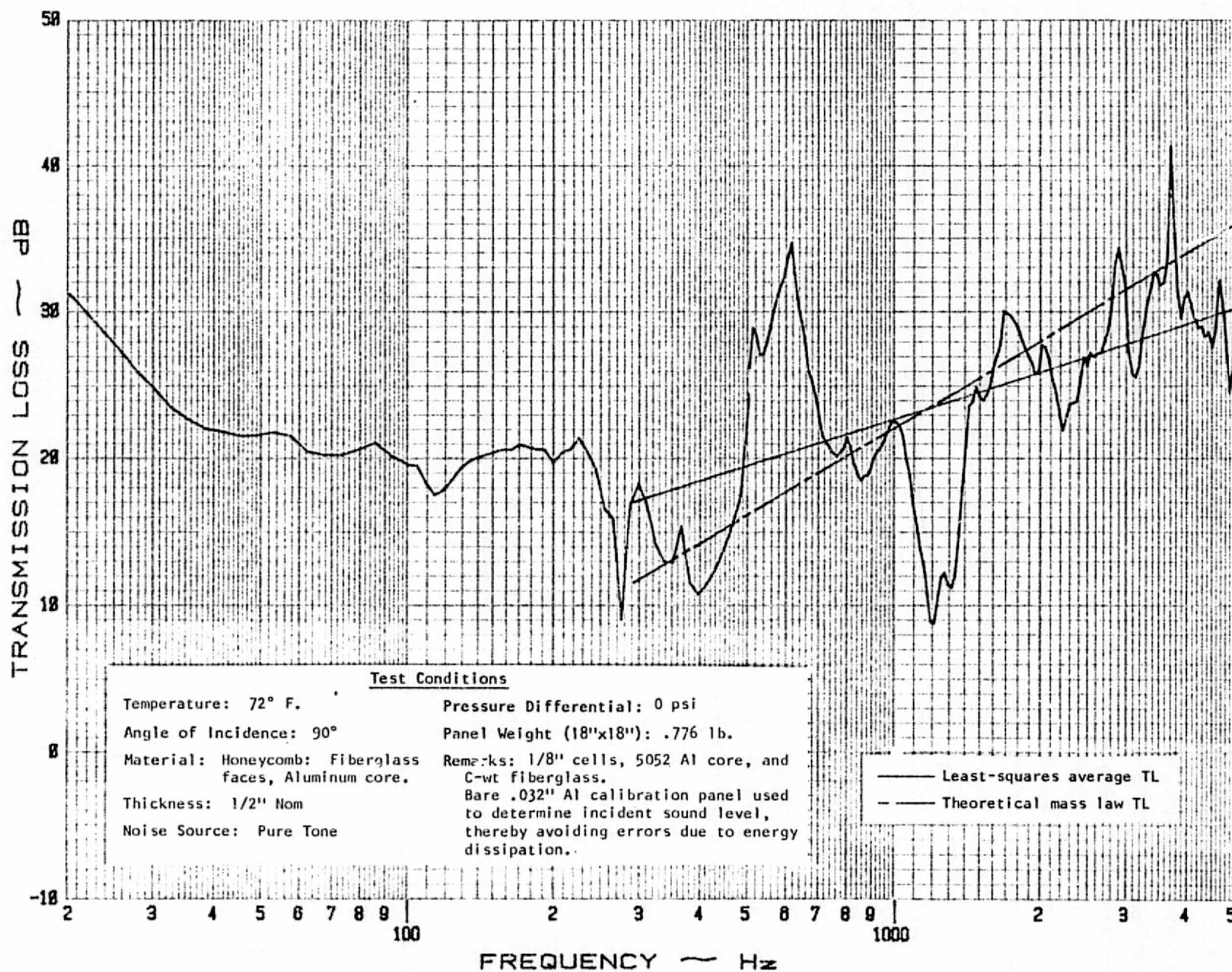


Figure B.32: Experimental Sound Transmission Loss of a Honeycomb Panel

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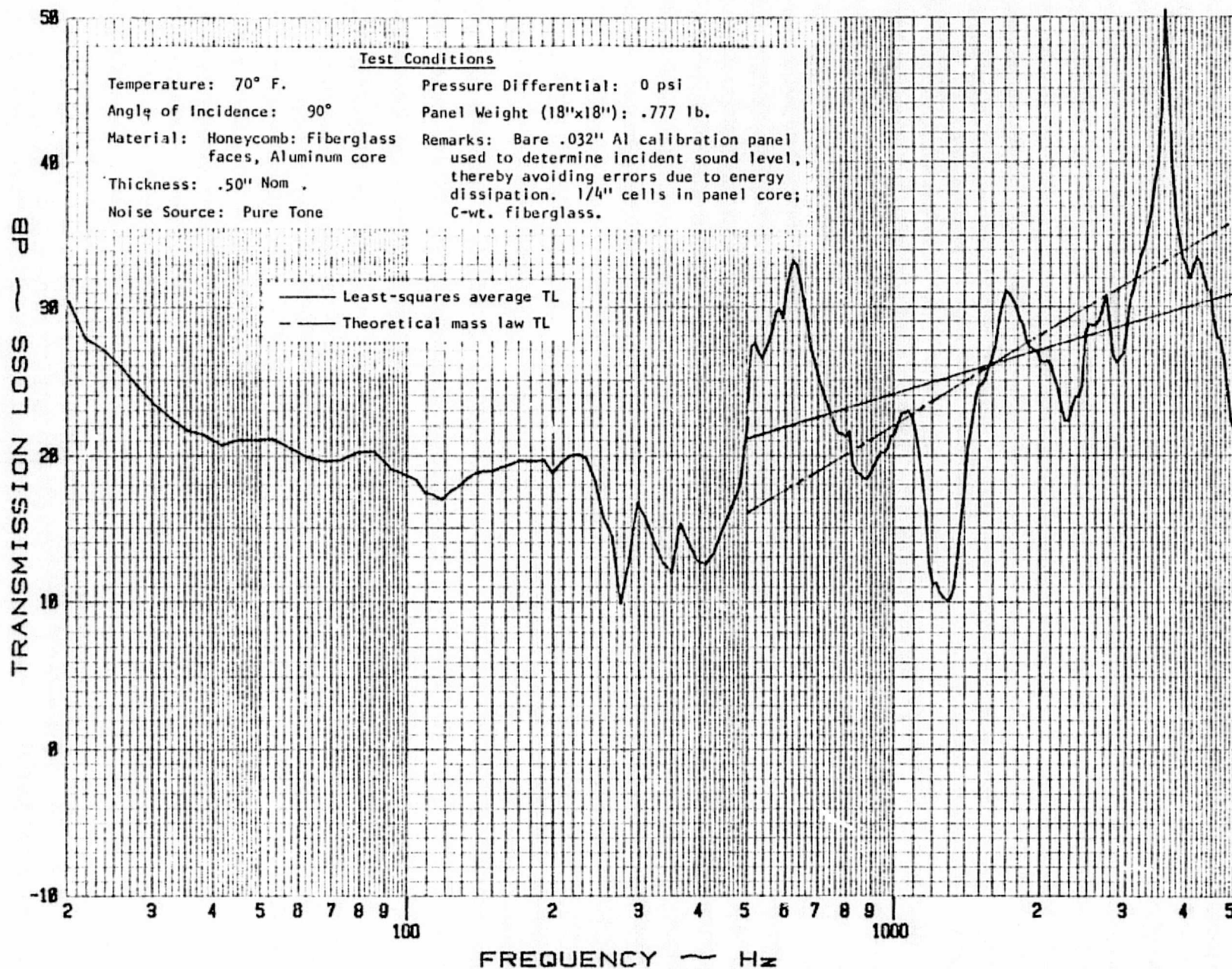


Figure B.33: Experimental Sound Transmission Loss of a Honeycomb Panel

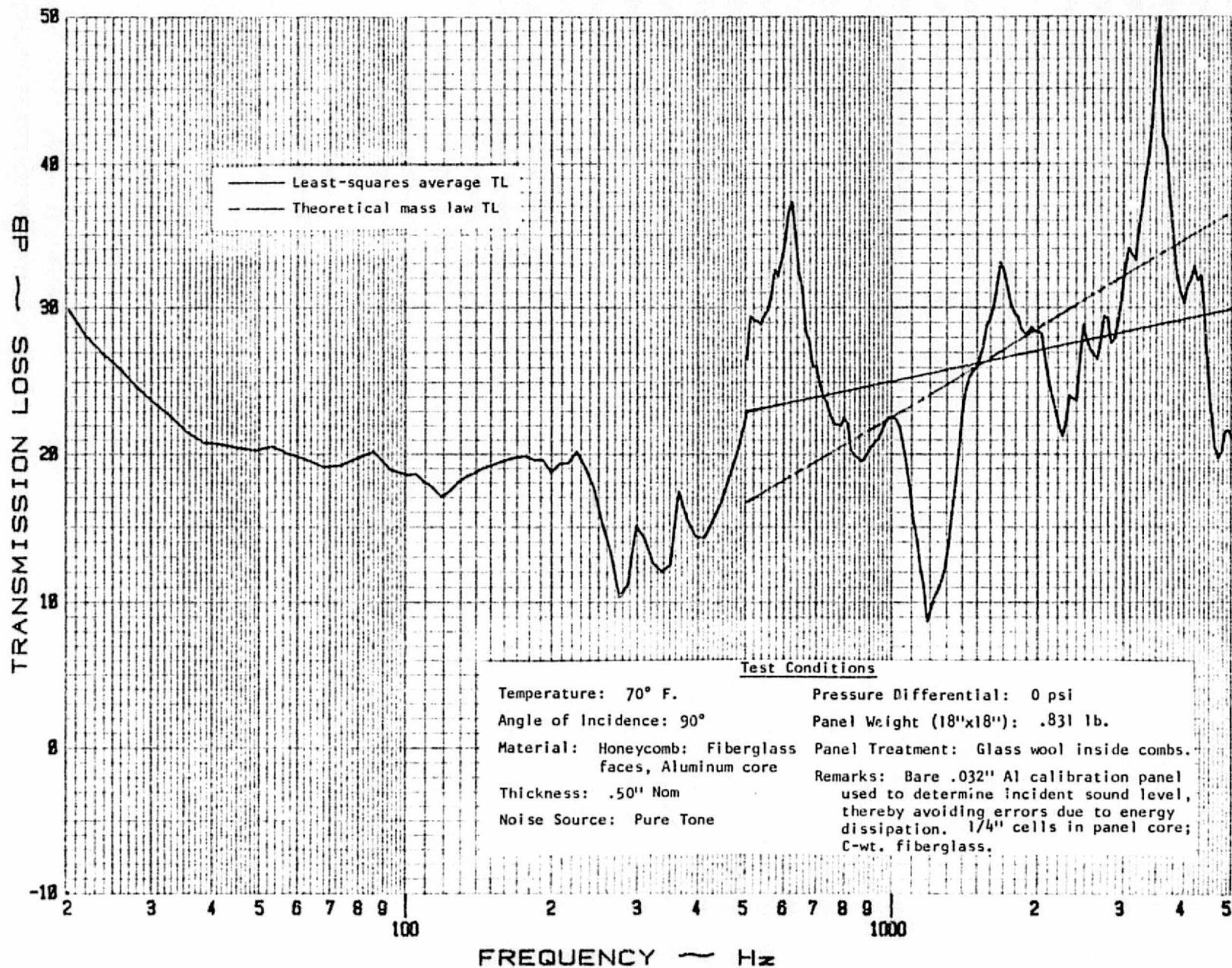


Figure B.34: Experimental Sound Transmission Loss of a Honeycomb Panel

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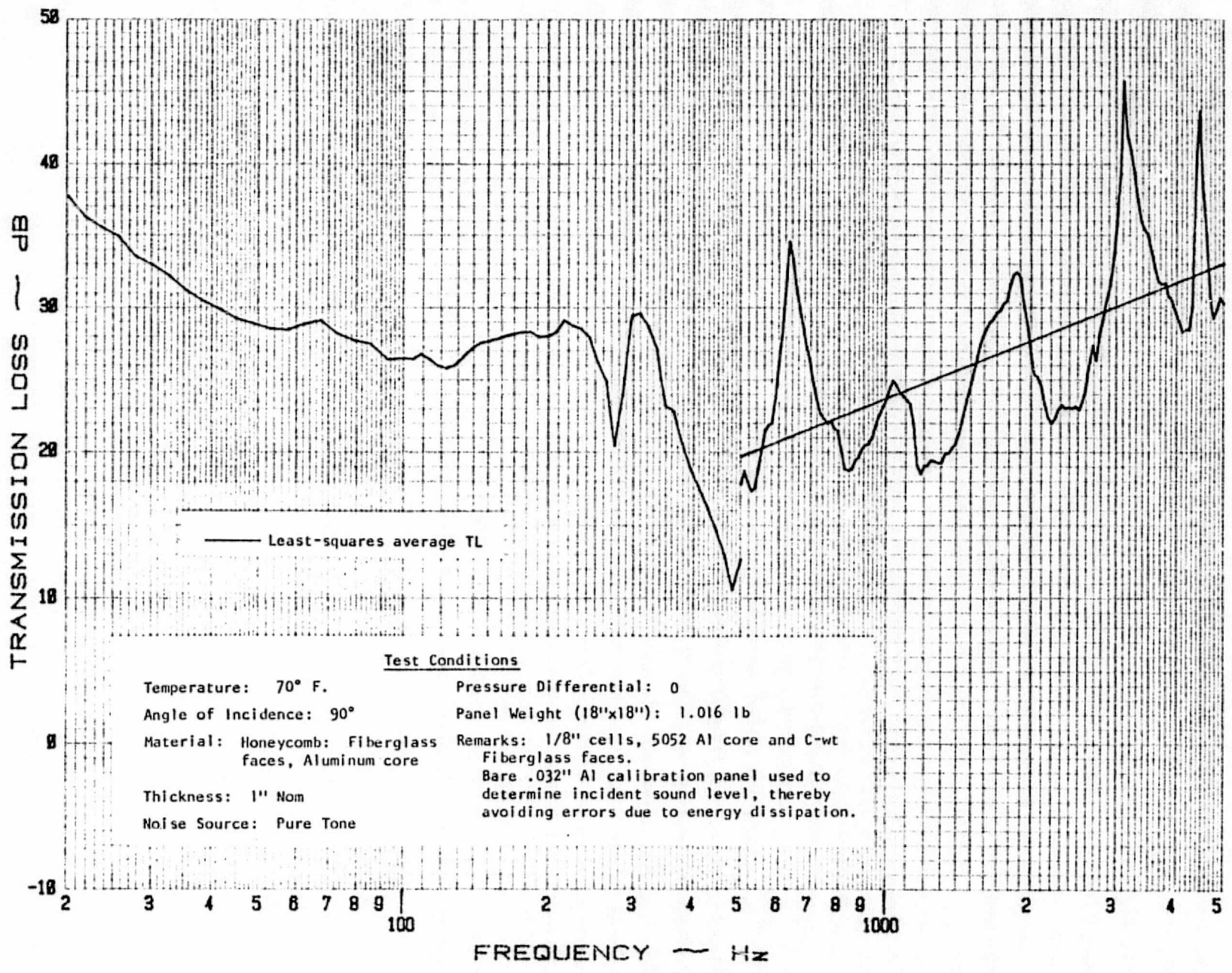
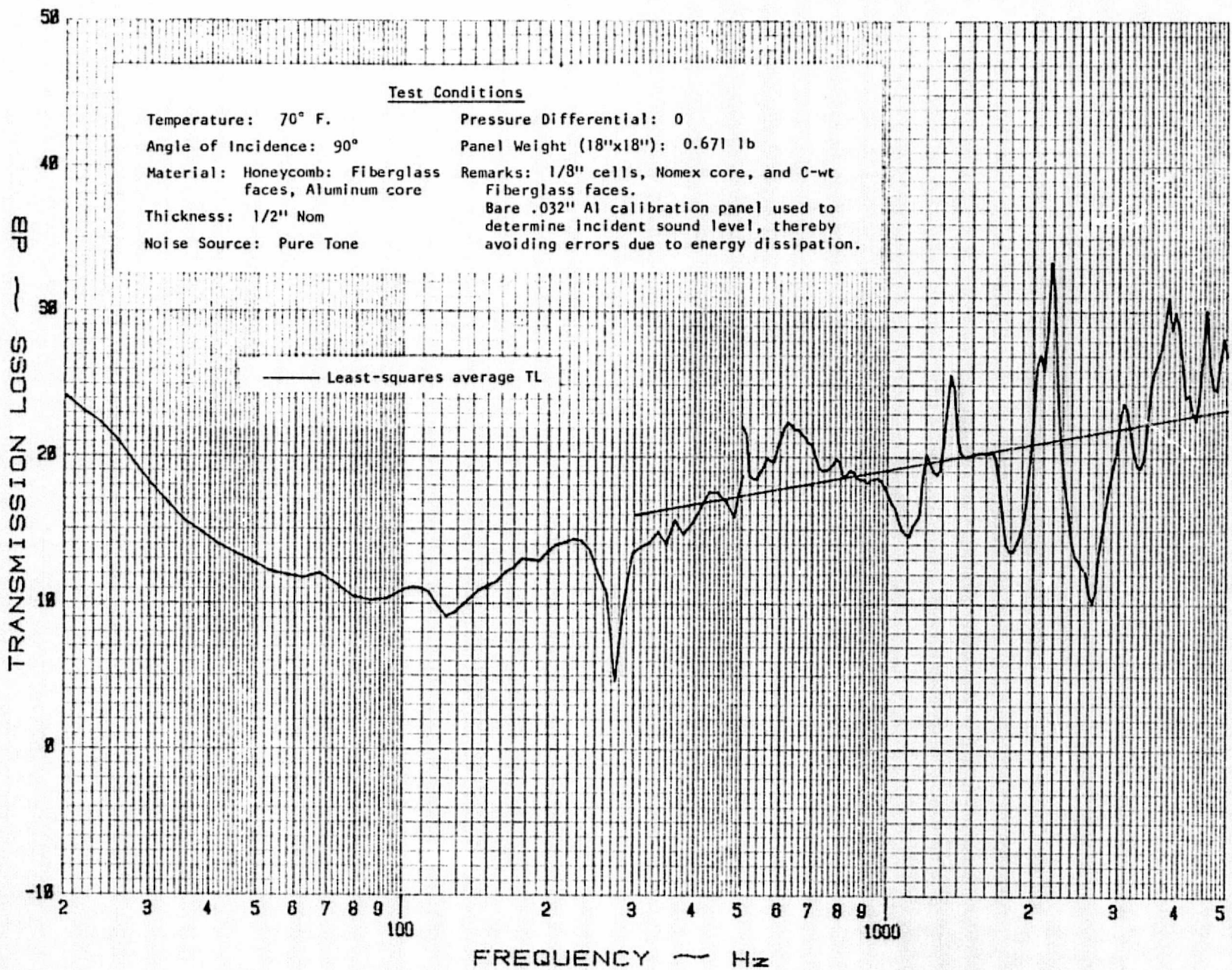
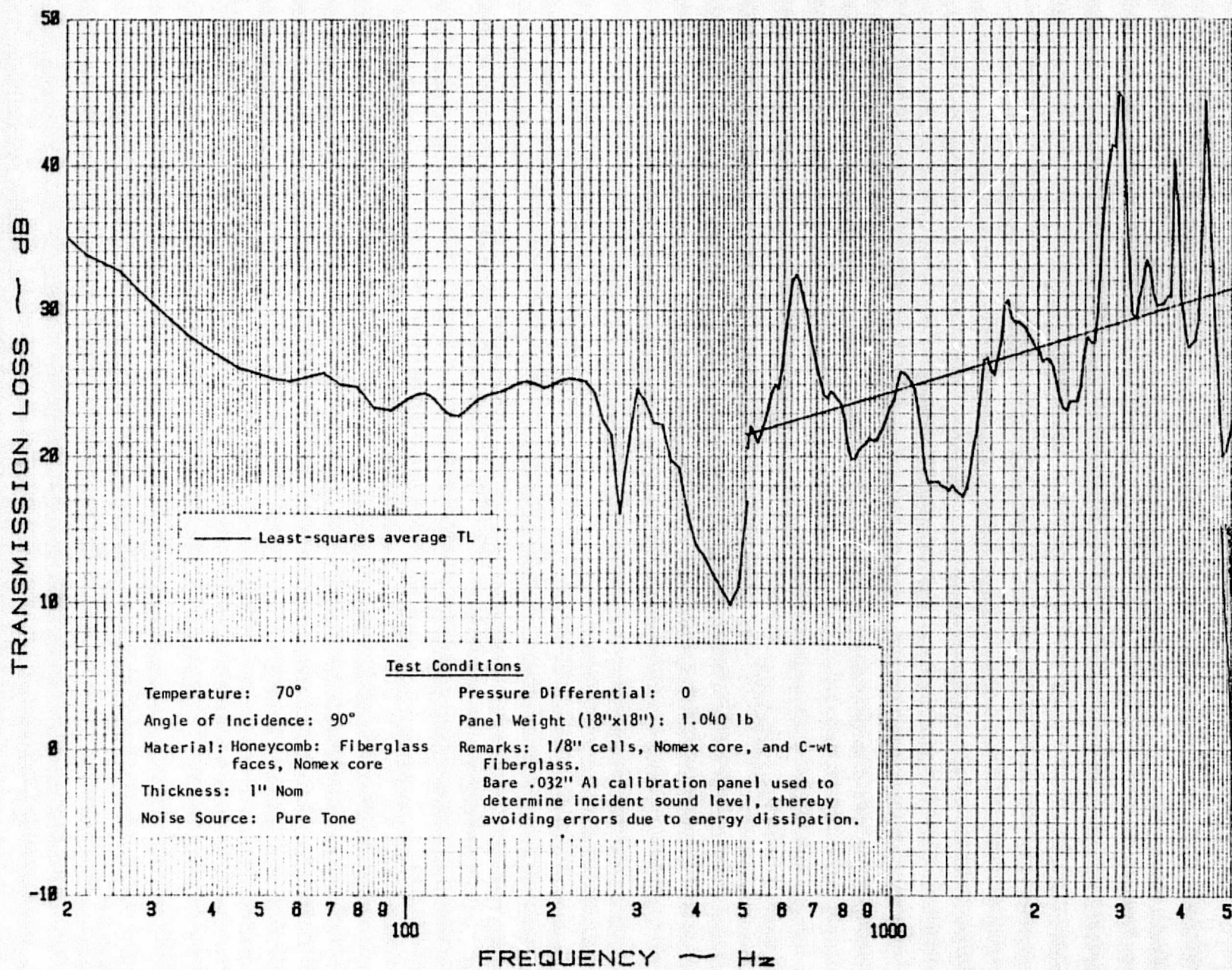


Figure B.35: Experimental Sound Transmission Loss of a Honeycomb Panel



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CALC		REVISED	DATE	Figure B.37: Experimental Sound Transmission Loss of a Honeycomb Panel	PAGE B.42
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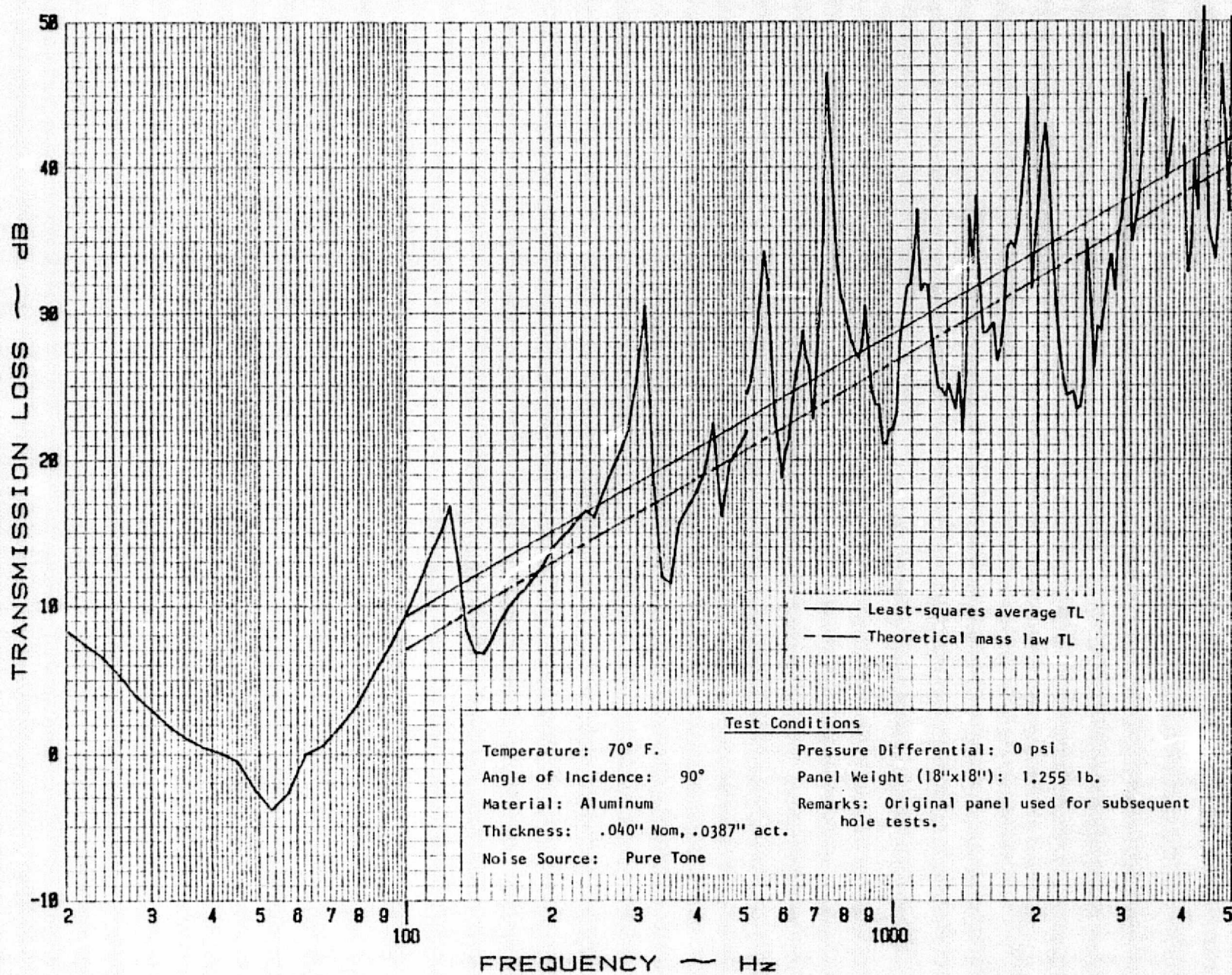


Figure B.38: Experimental Sound Transmission Loss of an Aluminum Panel

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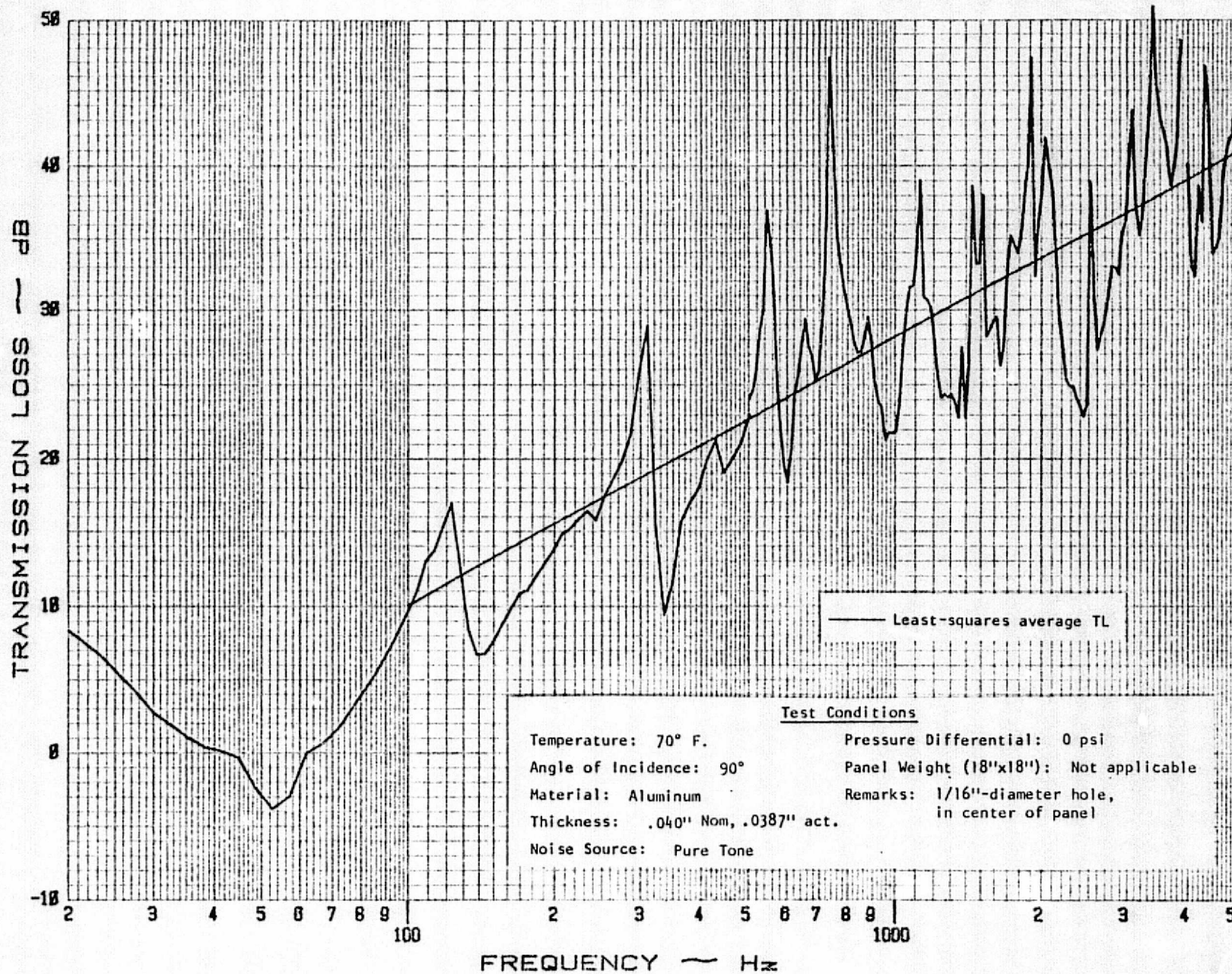


Figure B.39: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening

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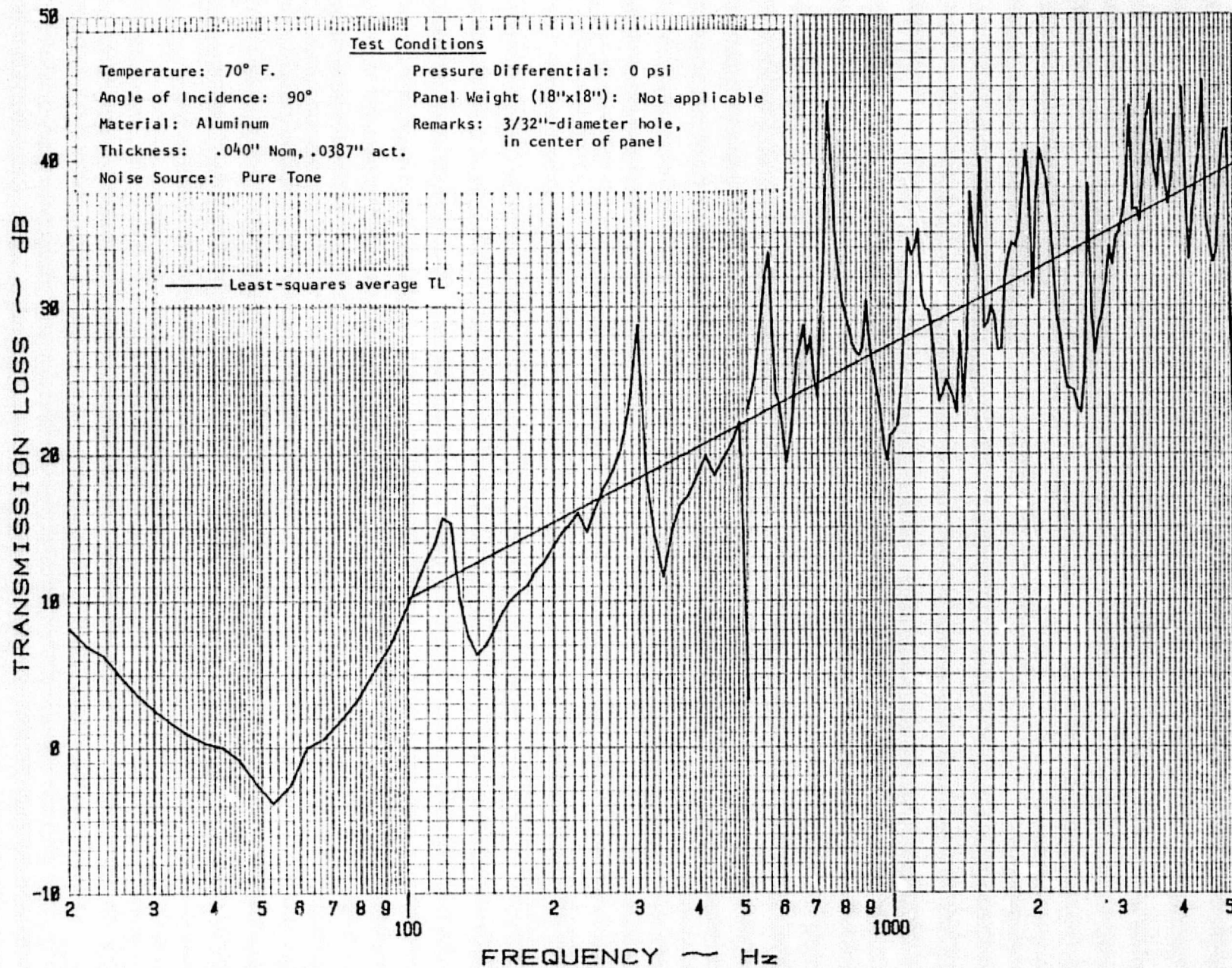


Figure B.40: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening

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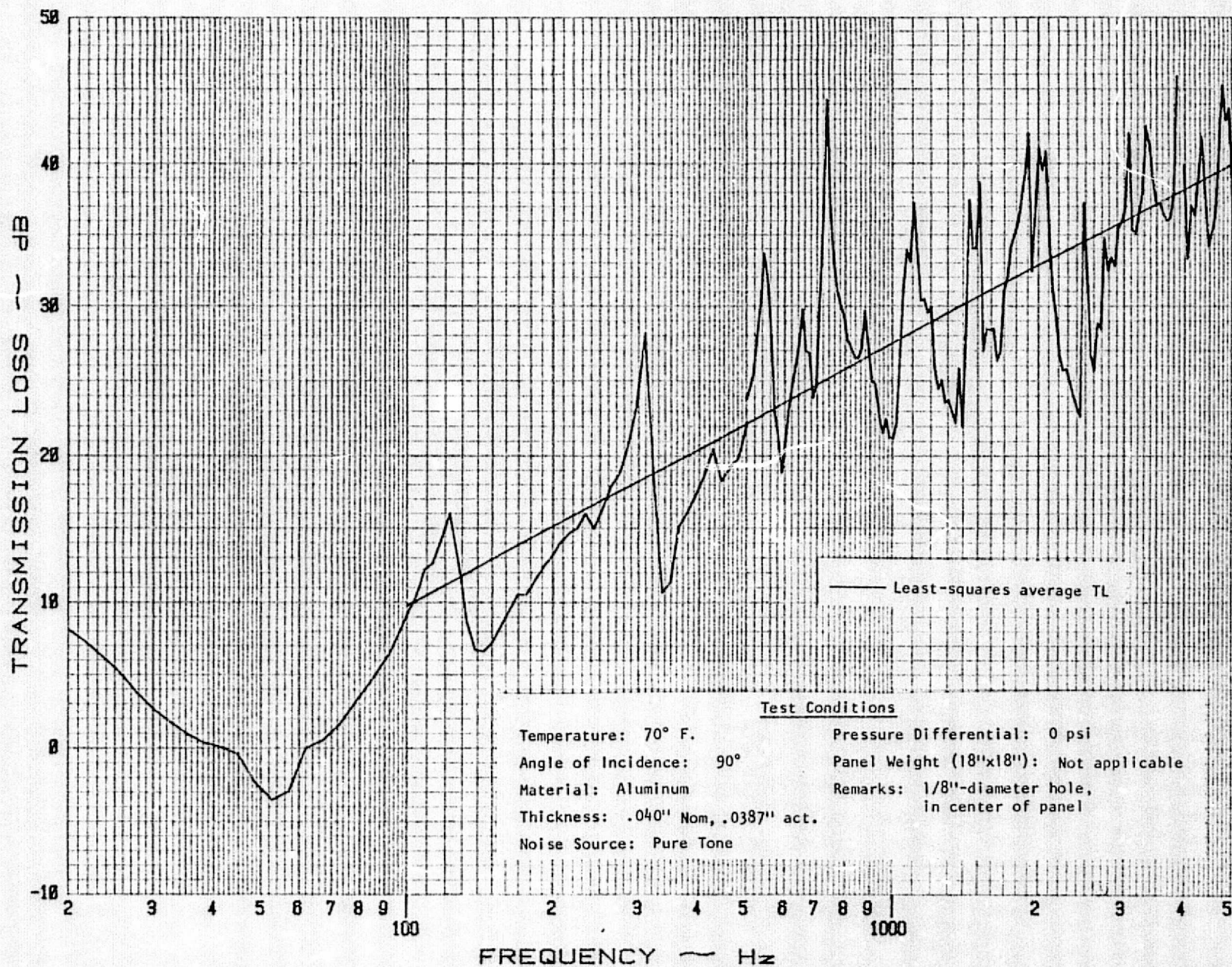
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CALC			REVISED	DATE	Figure B.41: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening	94
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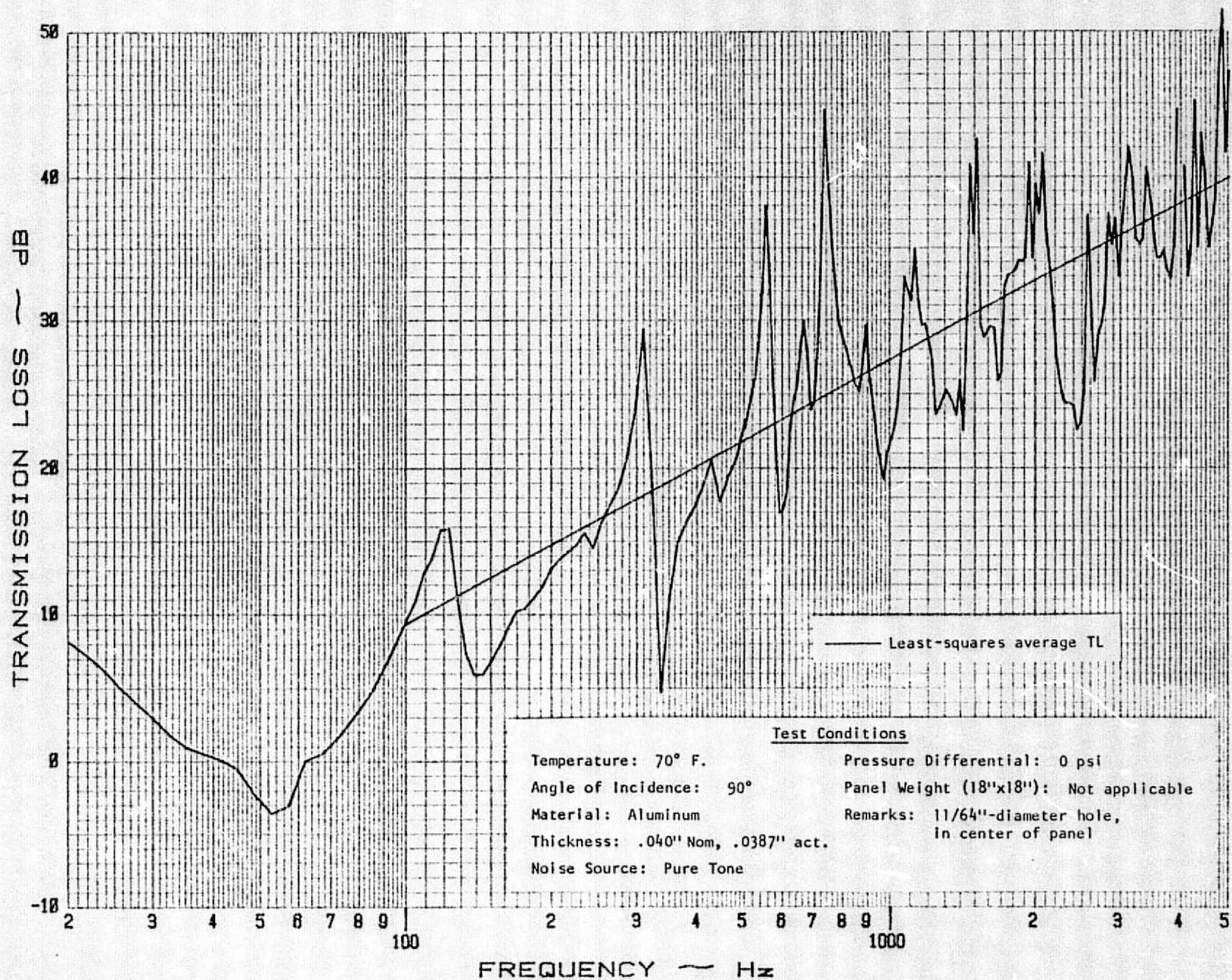


Figure B.42: Experimental Sound Trans-
mission Loss of an Aluminum Panel
with an Opening

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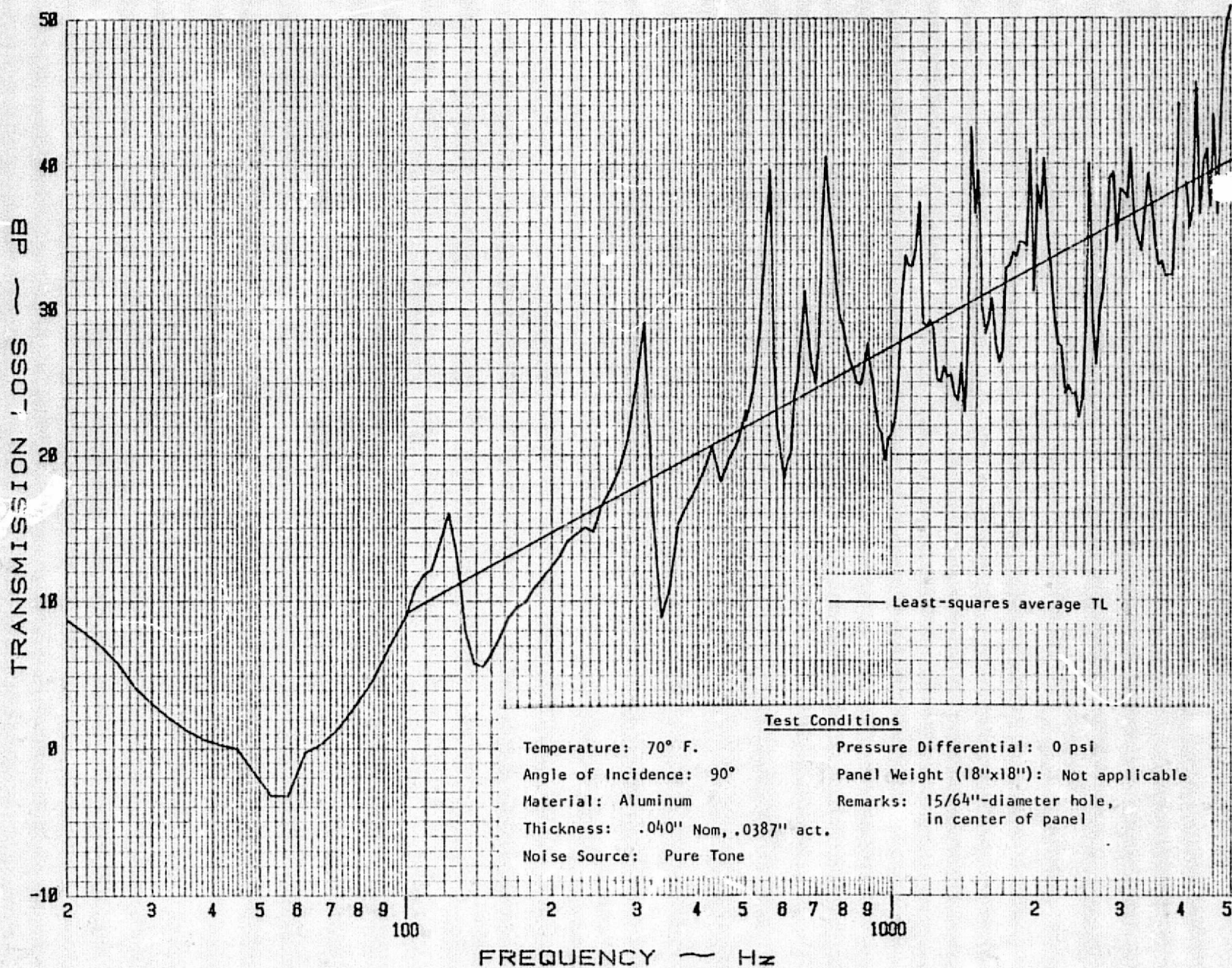
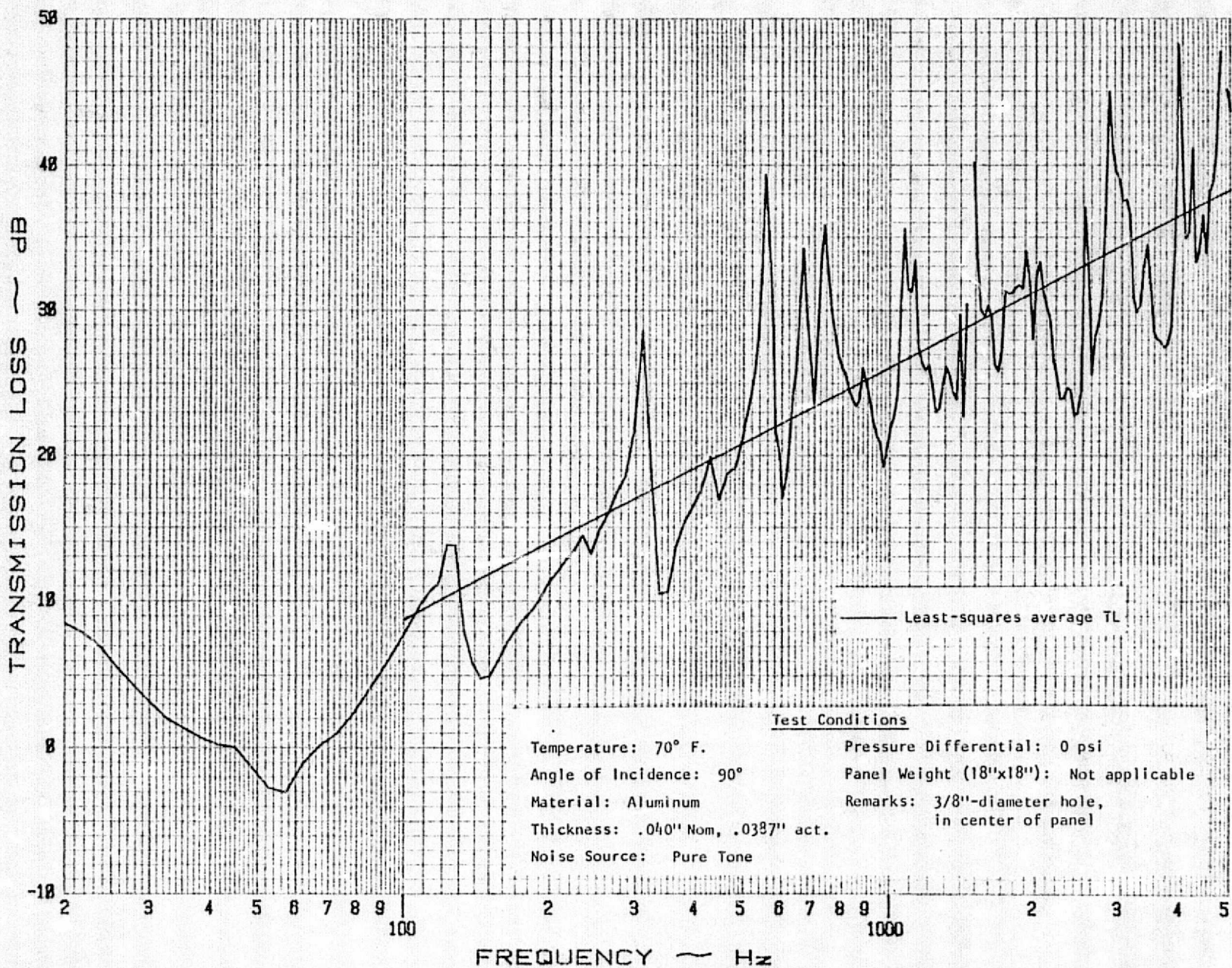


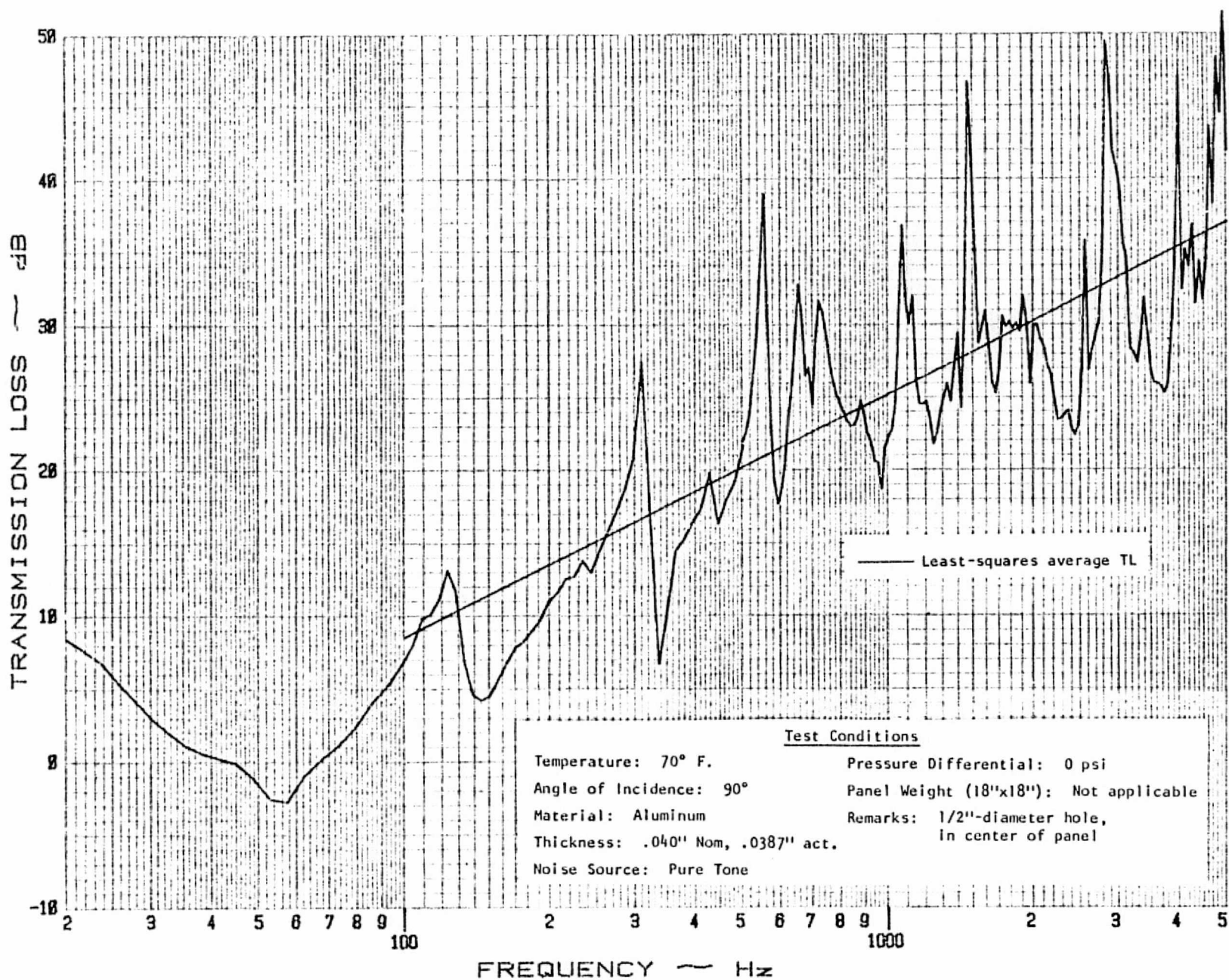
Figure B.4j: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening

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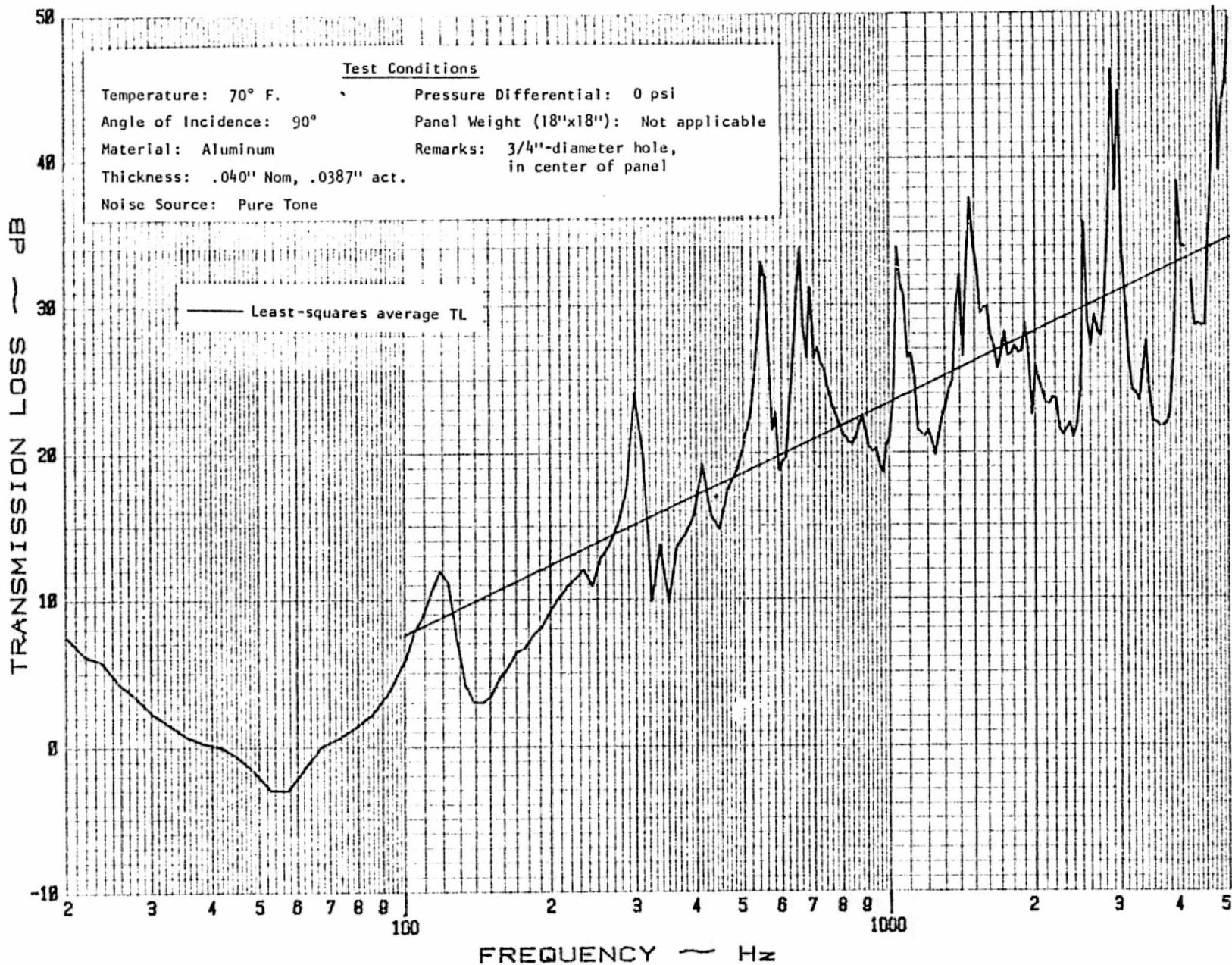
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CALC		REVISED	DATE	Figure B.44: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening	UNIVERSITY OF KANSAS	PAGE B.49
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CALC			REVISED	DATE	Figure B.46: Experimental Sound Trans- mission Loss of an Aluminum Panel with an Opening	101
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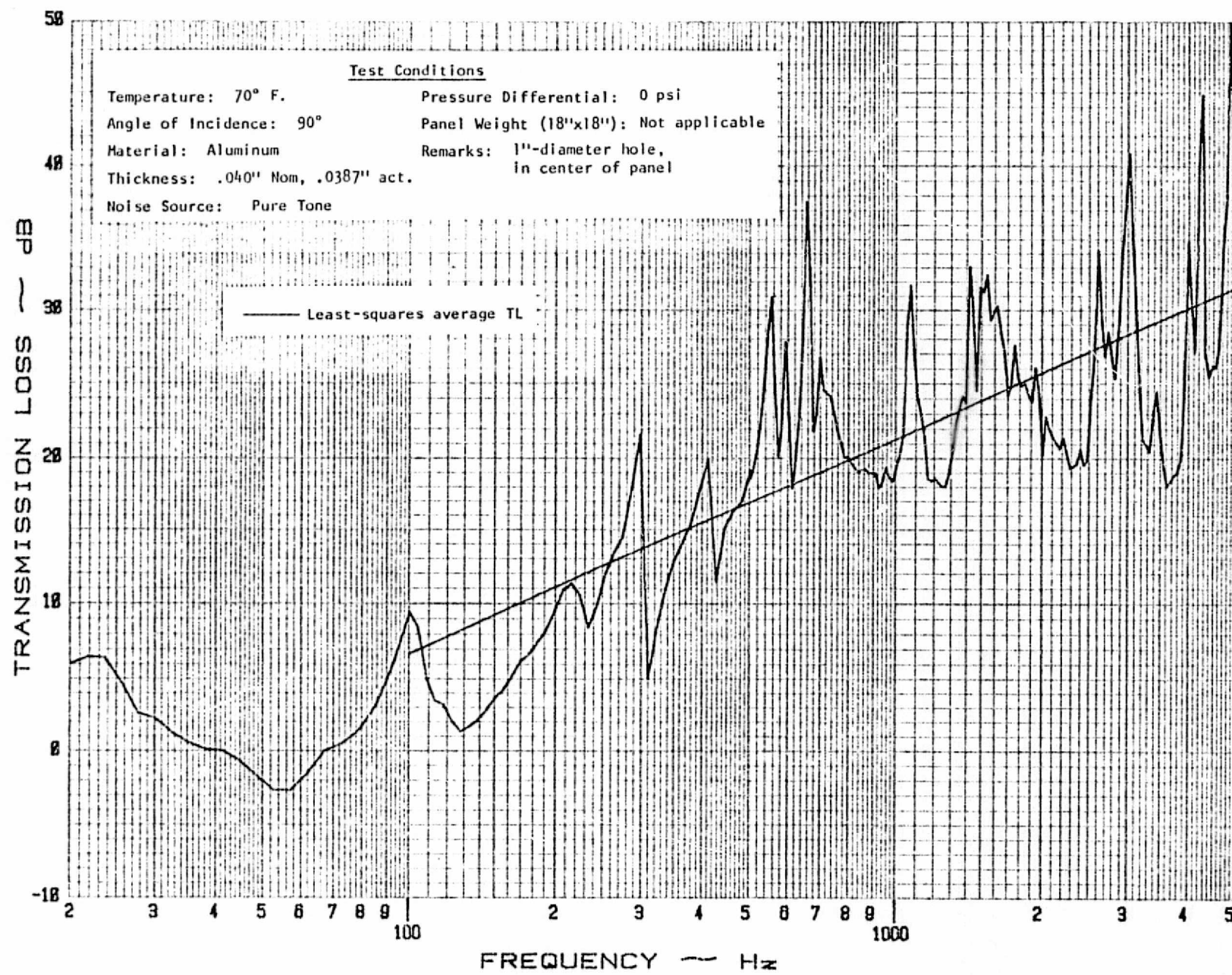


Figure B.47: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening

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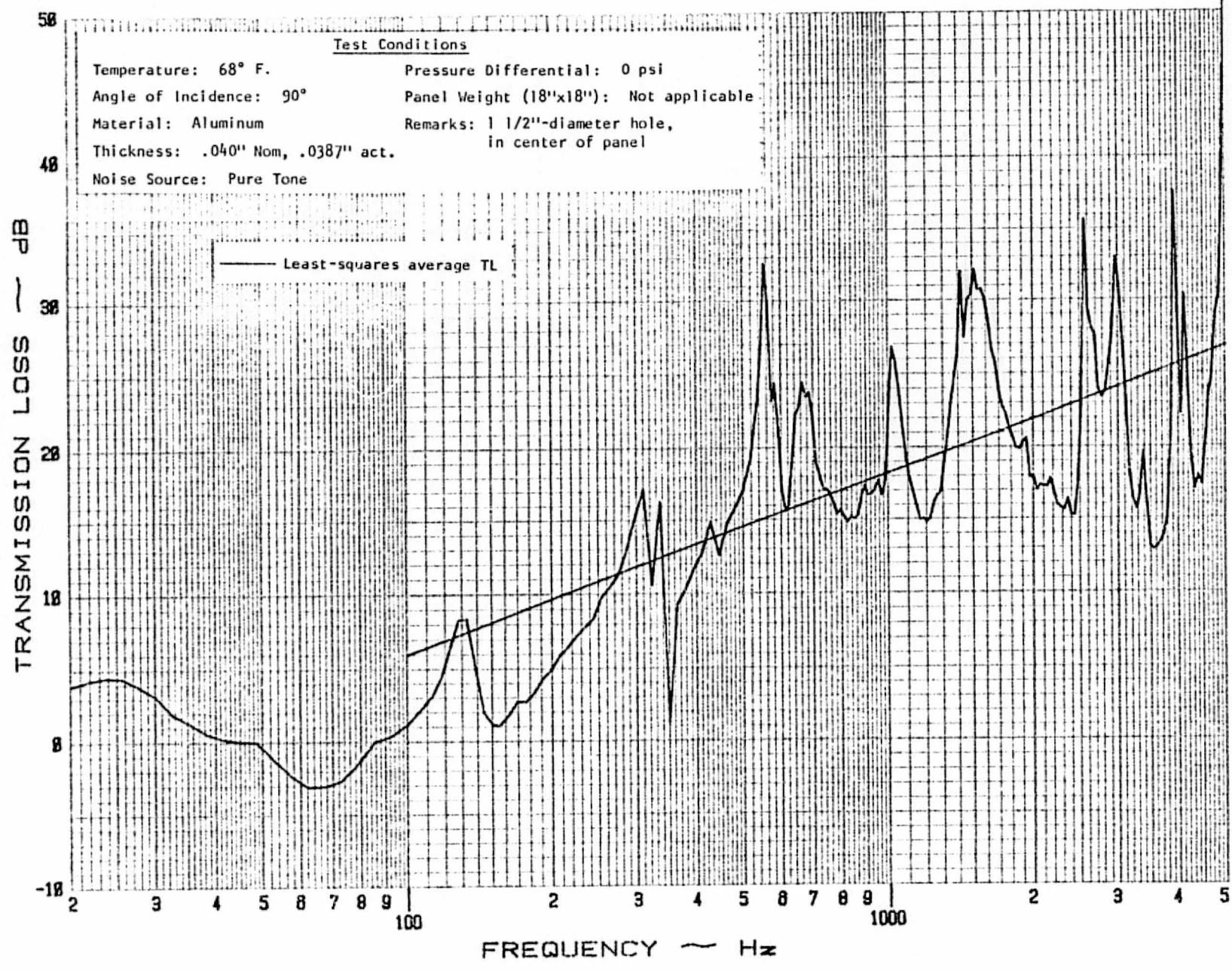


Figure B.48: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening

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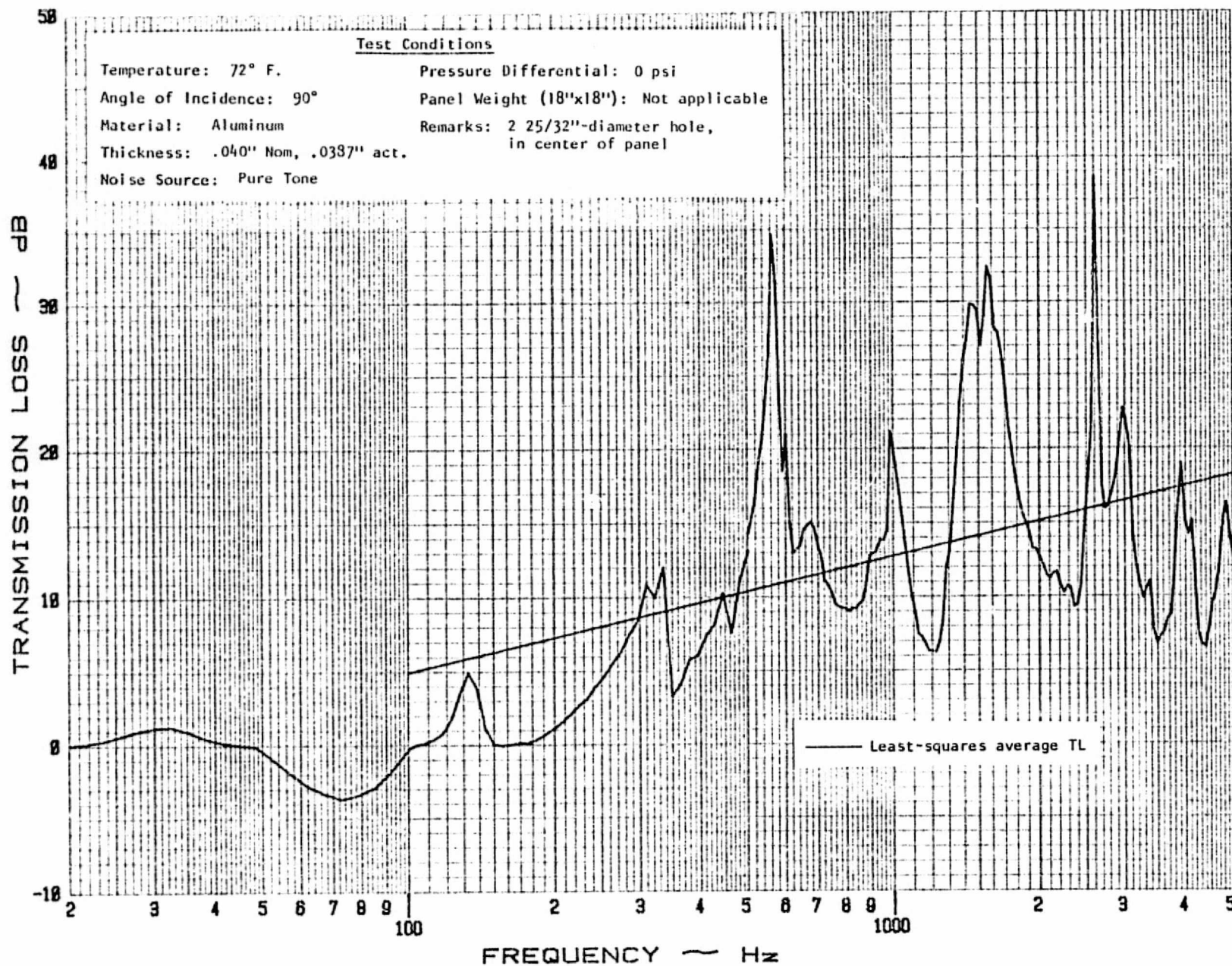


Figure B.49: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening

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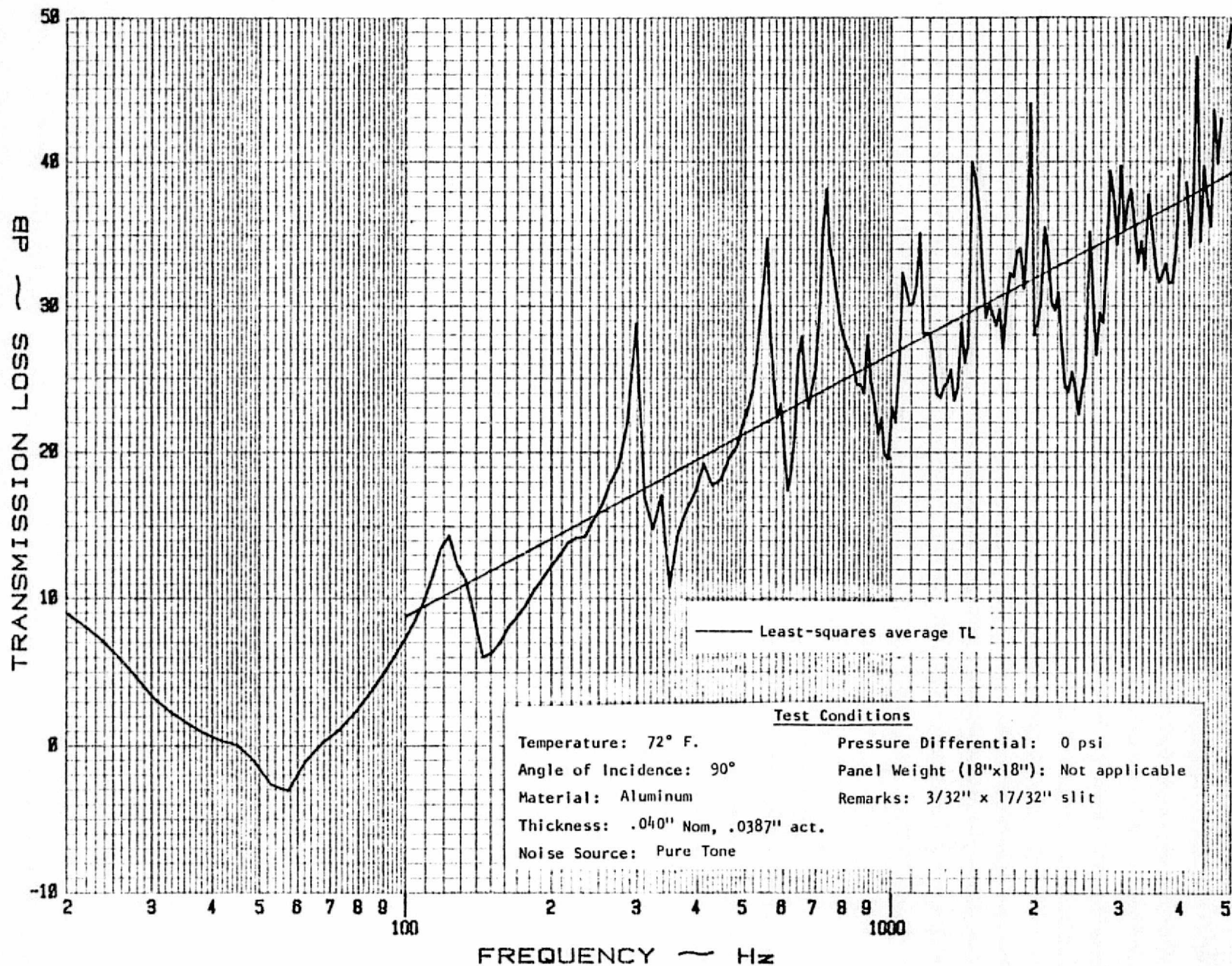


Figure B.50: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening

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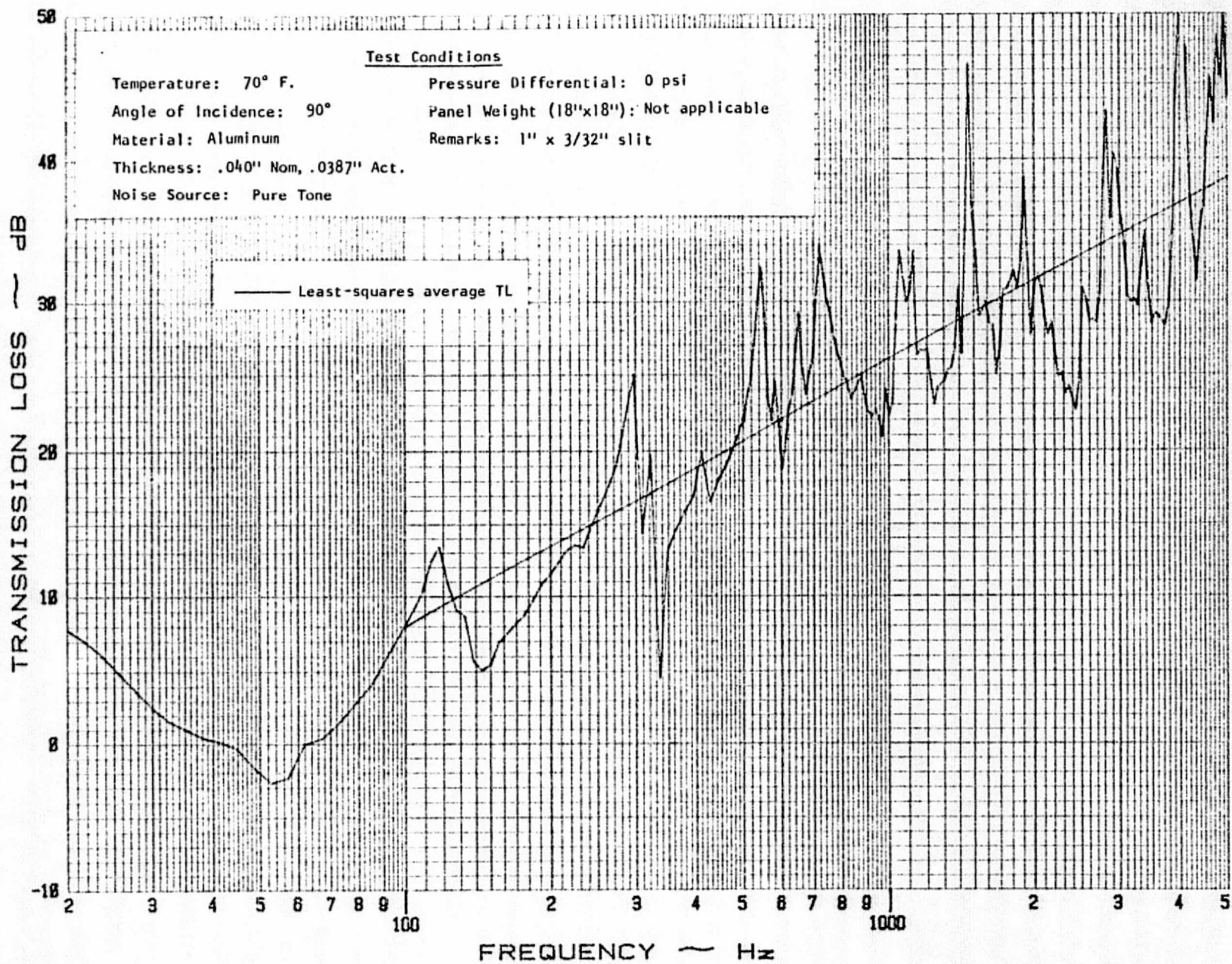


Figure B.51: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening

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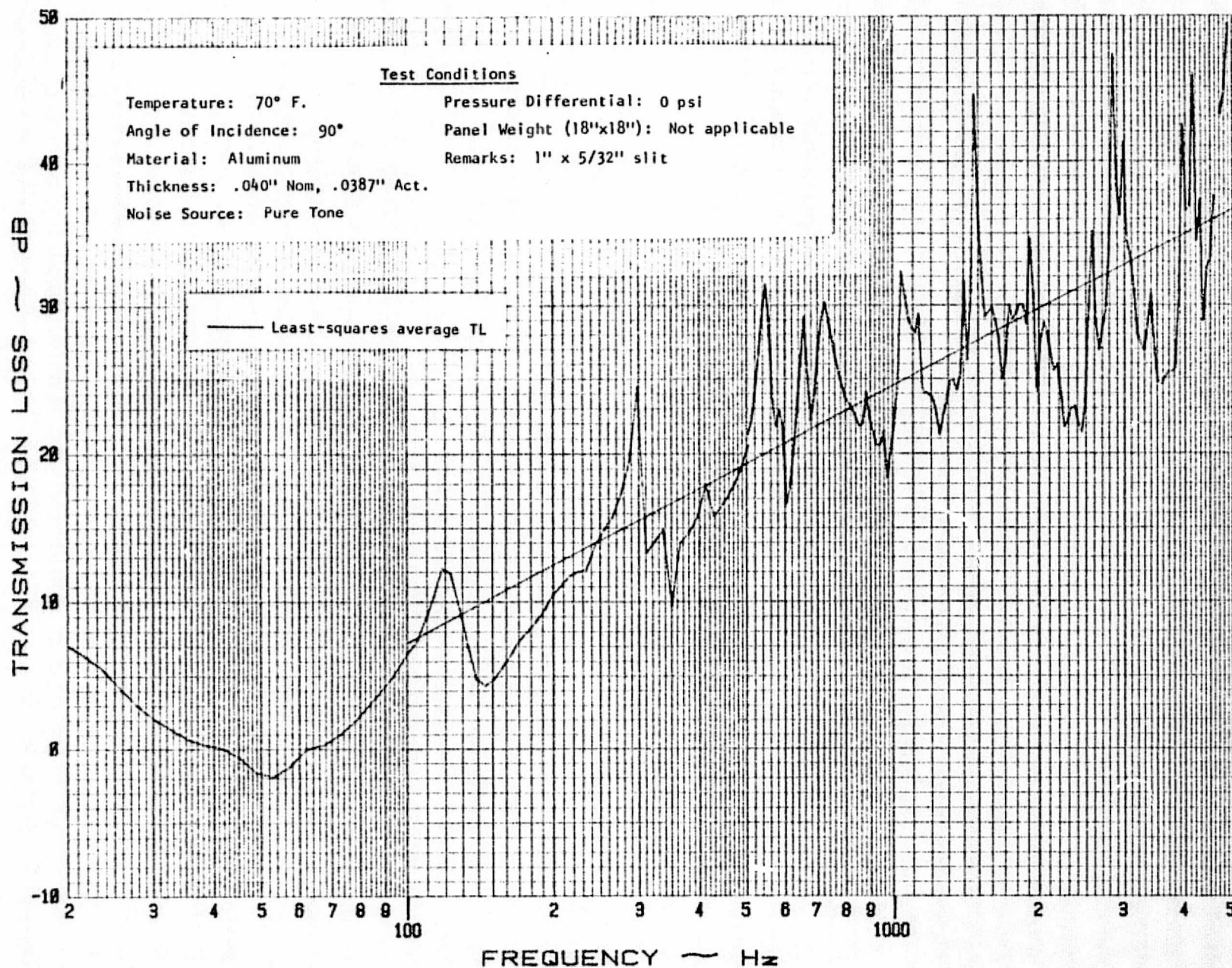


Figure B.52: Experimental Sound Transmission Loss of an Aluminum Panel with an Opening

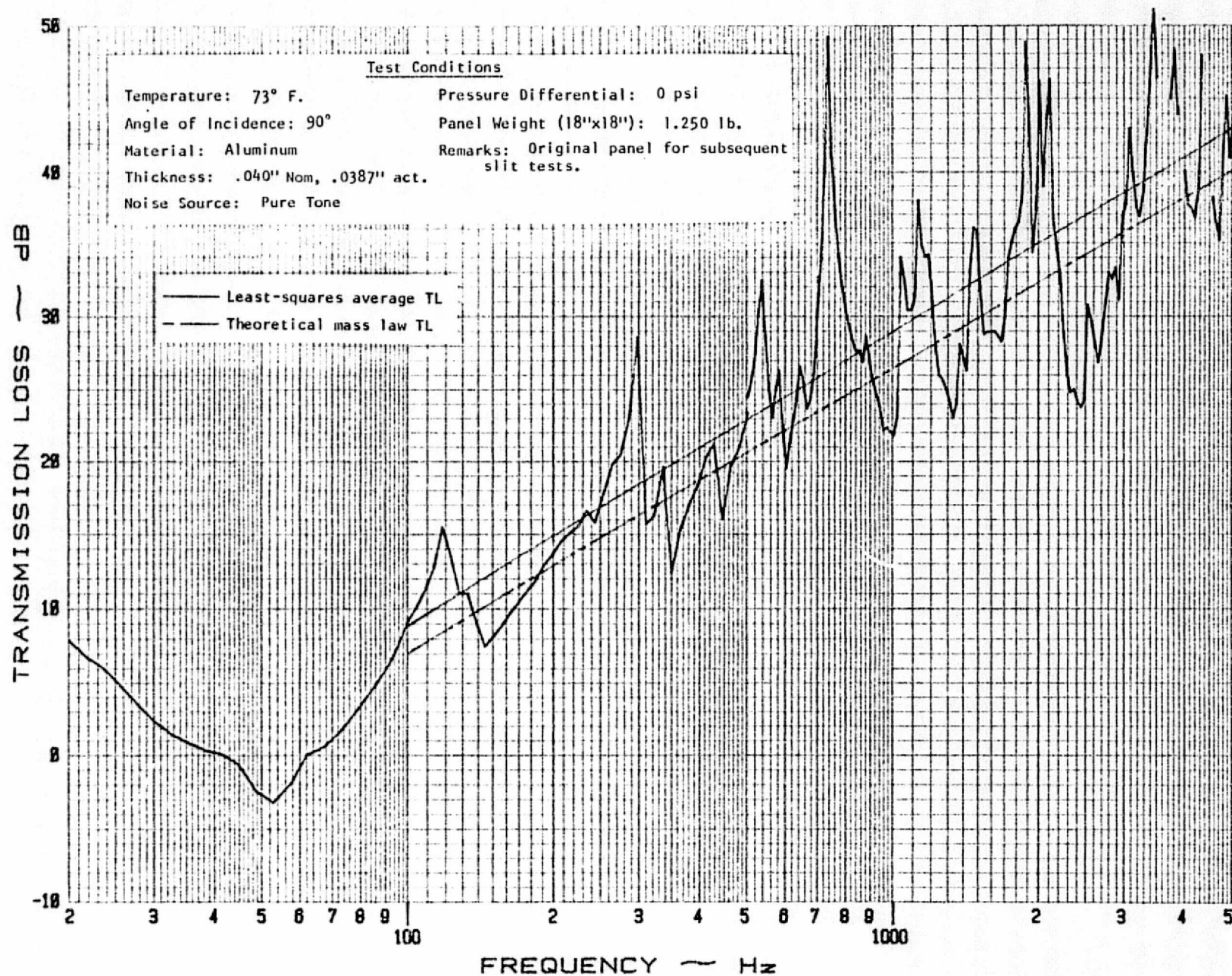


Figure B.53: Experimental Sound Transmission Loss of an Aluminum Panel

108

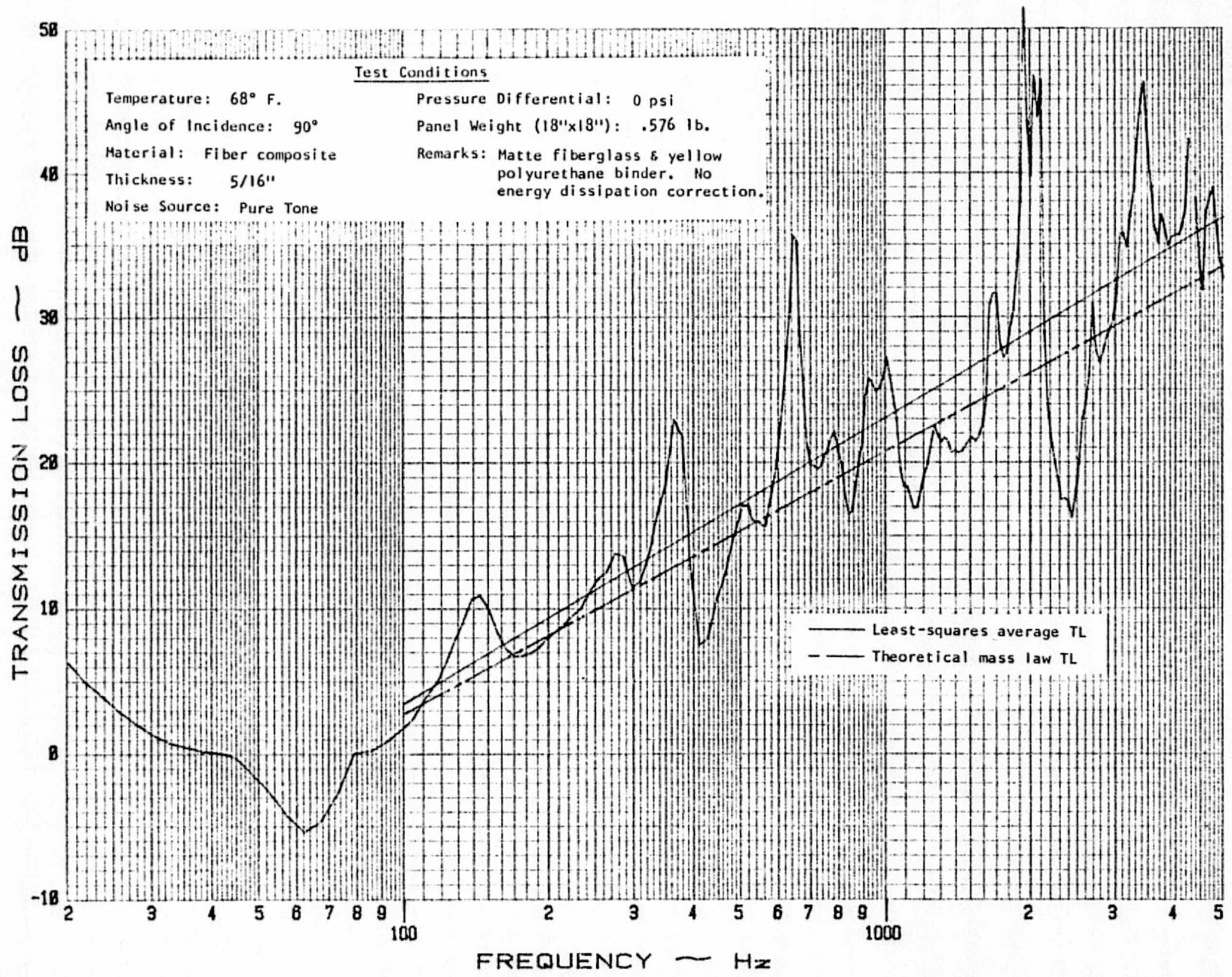


Figure B.54: Experimental Sound Transmission Loss of a Matte Fiberglass Panel

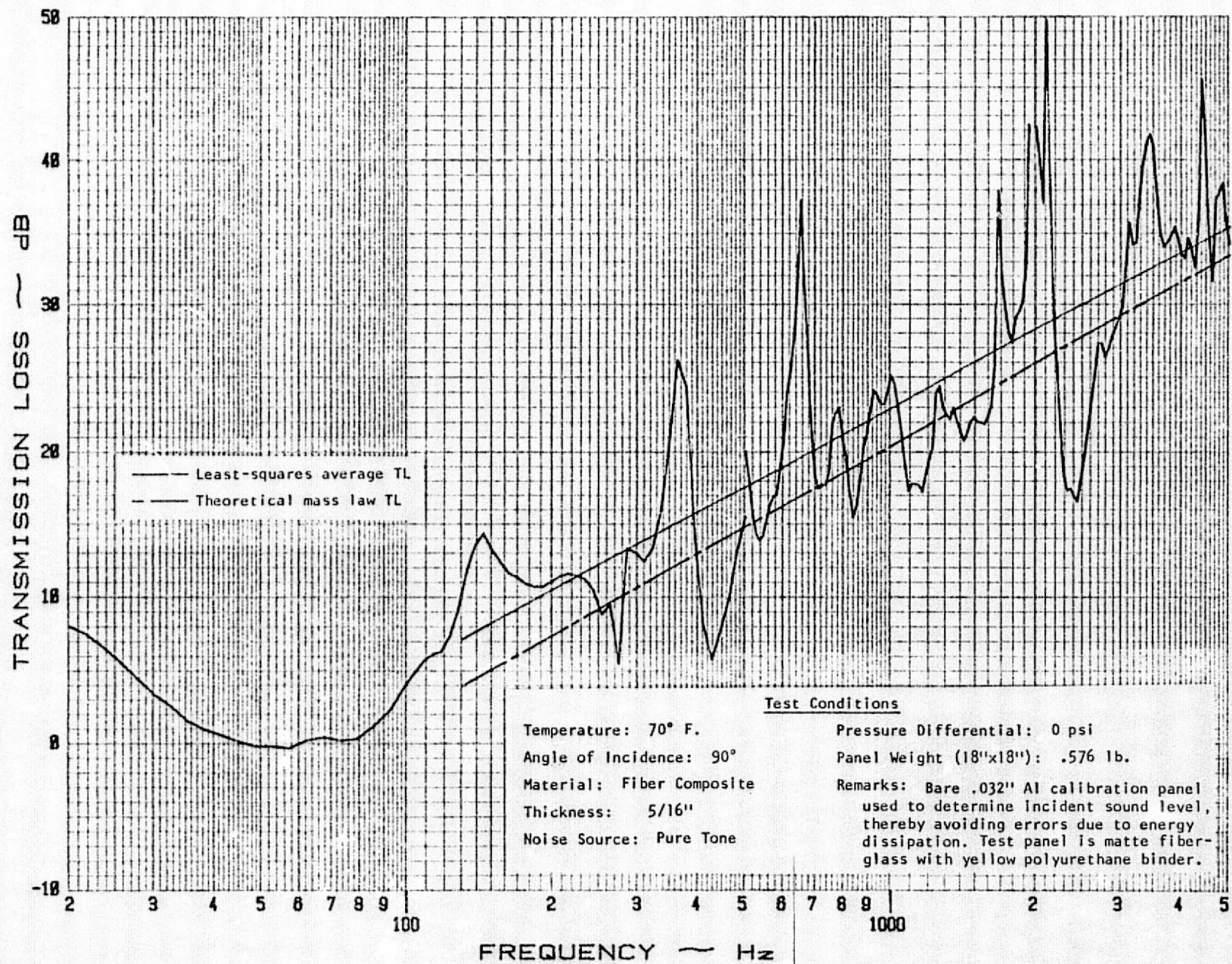


Figure B.55: Experimental Sound Transmission Loss of a Matte Fiberglass Panel

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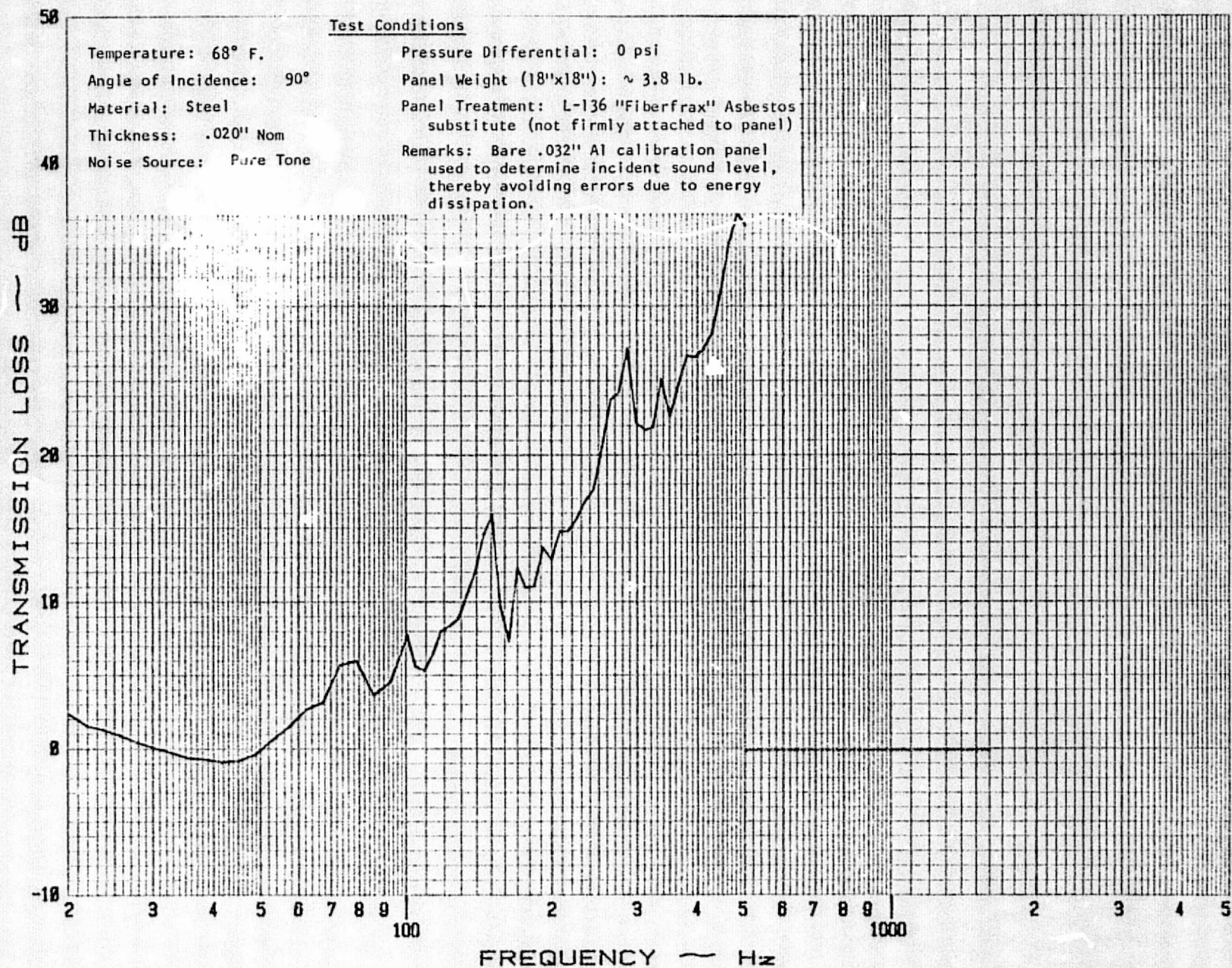


Figure B.56: Experimental Sound Transmission Loss of a Treated Steel Panel

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APPENDIX C
COMPUTER PROGRAM FOR
PREDICTION OF INSERTION LOSS

APPENDIX C

COMPUTER PROGRAM FOR PREDICTION OF PANEL INSERTION LOSS

This program was written in the language HPL for use with the Hewlett Packard 9825A Desktop Computer System. Program output is plotted automatically on an H.P. 9872A Plotter. The program is arranged to plot on semi-logarithmically ruled paper having 60 linear divisions on the y-axis. The horizontal logarithmic axis is assumed to span about $2 \frac{1}{3}$ cycles, specifically from 20 units to 5000 units. If other specifications of paper are used, the program must be altered accordingly. As written, the program sums Insertion Loss contributions from four cavity modes ($m,n = 0,2$) and sixteen panel modes ($q,r = 1,3,5,7$). Approximately six seconds are required to compute the IL at each frequency step. Frequency increments of 4 Hz are used; this results in a time requirement of about 25 minutes per case. This time could be shortened by specifying larger frequency increments or a narrower frequency range.

Initially, nine cavity modes ($m,n = 0,2,4$) were used. To reduce computation time, this was changed to four modes, as has been stated. No significant loss of accuracy was observed due to this change, but computation time per point dropped from about fourteen seconds to six.

Certain data are assigned values within the program. As a result, the program addresses a rather specific case. By referring to the flow chart, program listing and the variable list, a number of other cases could be accommodated. The program as listed pertains to a plain, flat aluminum panel backed by an absorption cavity of .457 m x .457 m (18 in. x 18 in.) cross section and of effective length as

described in Section 4.3 of this report. The panel edge conditions, for the purpose of determining in vacuo natural frequencies, are assumed to be clamped in the 1,1 mode, and midway between clamped and simply supported for all higher modes.

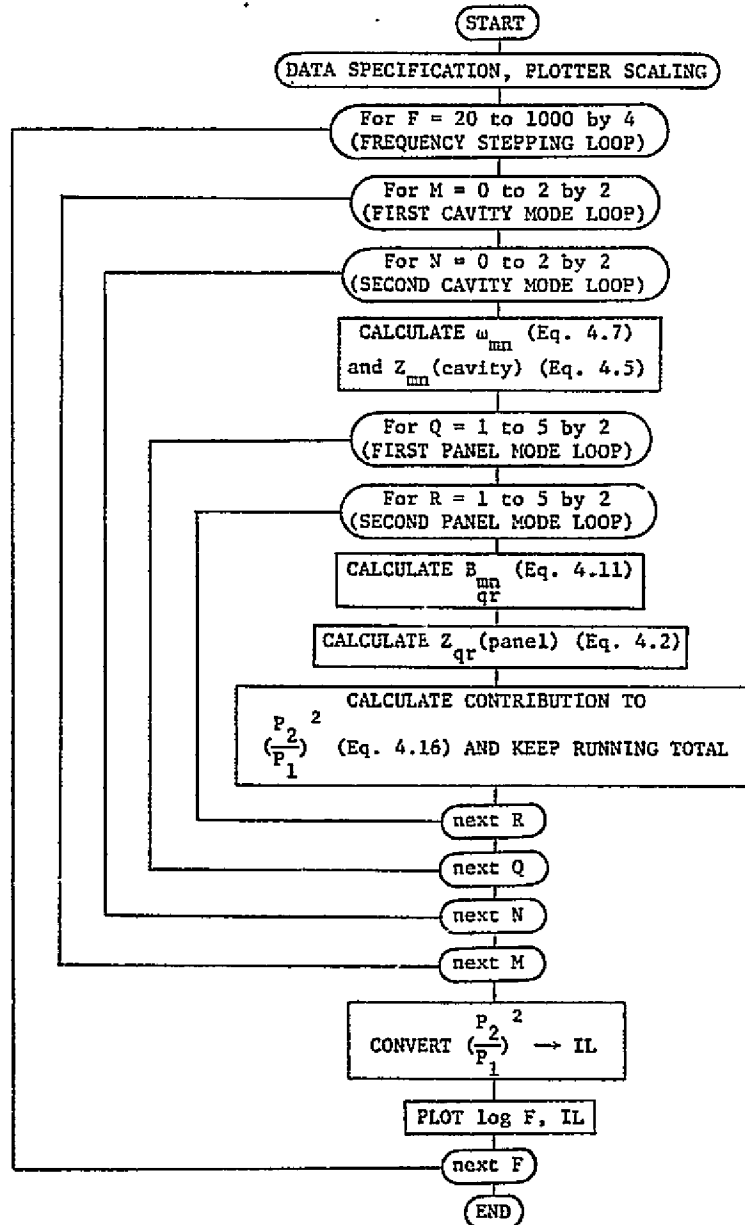
This case is specified in certain program lines. The elastic modulus, Poisson's ratio, and volume density of aluminum are given in lines 7-9. The cavity cross sectional area, $.209 \text{ m}^2$ is used in line 33. The effective length is specified in line 45. The non-dimensional panel frequencies (λ) of all odd,odd modes are given in lines 11-16 for the edge conditions which are described above and which presently appear to approximate those prevailing in the KU-FRL Facility. Frequencies for other conditions can be found in Table 4.1 and Figure 4.3.1.

The minimum distance between the microphone and the back wall of the cavity is given in lines 48 and 54. At the time this was written, it was noted that the values (.04 and .08 m) in these lines are not equal, as they should be. This discrepancy is not likely to affect results below about 500 Hz, but should be changed. A minimum value is needed because the expression for cavity length (incorrectly) yields lengths shorter than the microphone-panel distance at frequencies above roughly 400-500 Hz. As stated in Section 4.3, the expression for cavity length is probably incorrect at those frequencies, and needs improvement.

There are two additional assumptions which need to be stated. Cavity damping (sound absorption) is assumed to act in the $m = 0$, $n = 0$ cavity mode only. Line 43 orders a bypass of the cavity damping impedance calculation if either m or n is not zero. Likewise, panel

damping is assumed effective in the 1,1 mode only, and calculation of panel impedance due to damping is circumvented at line 66 for other modes. The final assumption is that the panel/cavity model used (Section 4.3) holds for both clamped and simply supported plates. While this assumption is probably not true, it does appear to be useful in the absence of a more exact model, as discussed in Section 4.3.

FLOW CHART FOR INSERTION LOSS PREDICTION PROGRAM
(HP 9825A)



IL PREDICTION PROGRAM

VARIABLES

Variable	Meaning	Units
A	Cavity length	m.
B	Panel/Cavity Coupling Coefficient	-
C	Speed of sound in air	m/s
D	Panel Stiffness $Eh^3/12(1-\nu^2)$	Nm
E	Modulus of Elasticity	N/m ²
F	Frequency	Hz
G	Local Barometric Pressure	mbar
H	Intermediate parameter $\alpha(t)$ always real calculation	-
I	Argument of <u>cosh</u> in Numerator, $(a-x)\alpha(t)$	-
J	$\cot(H)$ if $\alpha(t)$ is Im; $\coth(H)$ if $\alpha(t)$ is Re	-
K	Real Denominator of complex contribution to P_2/P_1	-
L	Edge Length of square panel	in, then m.
M	Cavity mode number	-
N	Cavity mode number	-
O	Mike distance from panel	m.
P	Magnitude of total complex P_2/P_1 at freq.	-
Q	Panel mode number	-
R	Panel mode number	-
S	Product of all P_2/P_1 complex contrib. terms except impedance	-
T	Panel thickness	in., then m.
U	First half of Coupling Coefficient calculation	-
V	Poisson's Ratio	-
W	Frequency ω	rad/sec
X	$\sqrt{\omega^2 - \omega_{mn}^2}$ or $\sqrt{\omega_{mn}^2 - \omega^2}$, whichever is real	1/sec

IL PREDICTION PROGRAM

VARIABLES (continued)

Variable	Meaning	Units
Y	Decision logic variable: 1 = print panel nat. freqs.	-
Z	Dimensionless panel damping factor	-
A[1]	$\rho_{\text{air}} c^2 \cdot (\text{tube cross sect. area})$	
A[2]	Acoustic resistance of cavity abs. material	rayls
A[3]	Cavity volume (effective)	m ³
A[4]	Acoustic capacitance of cavity (Olson)	m ³ /N
D[1]-D[20]	X-coordinates of logarithmic scale digits	-
K[1]	K of Guy & Bhatta.'s P_2/P_1 equation	-
K[2]	K' " " " " " "	-
K[3]	$(16/\pi^2 q r)$ of Guy & Bhatta.'s P_2/P_1 equation	-
L[1,1]-[4,4]	Non-dim odd, odd nat. freqs of clamped square panels, λ_{qr}	-
L[1,4]	[example:] non-dim nat. freq of 1,7 mode	-
O[1,1]-O[4,4]	Dimensional odd, odd, nat. freqs of cl. sq. panels, ω_{qr}	rad/sec
P[1]	Real part of P_2/P_1 , for a given frequency	-
P[2]	Imag part of P_2/P_1 , for a given frequency	-
R[1]	Panel material density, ρ	kg/m ³
R[2]	Air density at 20°C and specified baro. press.	kg/m ³
W[1,1]-W[3,3]	Frequencies of even, even cavity modes	rad/sec
Y[1] & Y[2]	cosh I and cosh H, resp., if args real; cos(I) and cos(H) if imag.	
Z[1,1] & Z[1,2]	Real and Imag parts of cavity impedance	rayls
Z[2,1] & Z[2,2]	Real and Imag parts of panel impedance	rayls


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0: "IL Prediction by Impedance Method":
1: dim R[2]
2: scl 1.30103,3.69897,-10,50
3: wrt 705,"VR"
4: lim 1,3.8,-10,50
5: ent "Panel thickness, in.",T
6: T/39.37+T
7: 7.24e10+E
8: .33+V
9: 2768+R[2]
10: E*T/3/12(1-VV)+0
11: dim L[4,4]; dim O[4,4]; dim W[3,3]
12: 35.984+L[1,1];198.77+L[2,2]
13: 115.16+L[1,2]+L[2,1];292.87+L[1,3]+L[3,1]
14: 364.51+L[2,3]+L[3,2];529.48+L[1,4]+L[4,1]
15: 610+L[2,4]+L[4,2];770.43+L[3,4]+L[4,3]
16: 527.48+L[3,3];1014.38+L[4,4]
17: ent "Edge length of sq. panel, in.",L
18: L/39.37+L
19: ent "dimensionless damping",Z
20: ent "Mike distance from panel, m.",C
21: for Q=1 to 4
22: for R=1 to 4
23: (L[Q,R]/LL)*r(D/R[2]T)+O[Q,R]
24: next R
25: next Q
26: 343+C
27: rad
28: dim Z[2,2]; dim Y[2]; dim P[2]
29: dim K[3]
30: ent "local bore, mbar",G
31: 1.2006G/1013+R[1]
32: dim A[4]
33: R[1]CC=.209+A[1]
34: ent "cav res, rays",A[2]
35: ent "cav vol, (m)3",A[3]

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36: A[3]/A[1]+A[4]
37: pen# 1
38: for F=20 to 1000 by 4
39: 2πF+W;0+P[1];0+P[2]
40: for M=0 to 2 by 2
41: for N=0 to 2 by 2
42: πCf(MM/LL+NN/LL)+W[(M+2)/2,(N+2)/2]
43: if M or N;sto +2
44: A[2]/(1+WWA[2]+2A[4]+2)+Z[1,1]
45: 1.868-.2904ln(F)+A; if F<130;31.172*F+(-.8685)+A
46: if W[(M+2)/2,(N+2)/2]>W;sto "REAL"
47: r(WW-W[(M+2)/2,(N+2)/2]+2)+X;AX/C+H
48: max(A-0,.04)X/C+I
49: cos(H)/sin(H)+J;sf# 14
50: -JR[1]CW/X+Z[1,2];cf# 14
51: cos(H)+Y[2];cos(I)+Y[1]
52: sto "cavdone"
53: "REAL":r(W[(M+2)/2,(N+2)/2]+2-WW)+X;AX/C+H
54: max(A-0,.08)X/C+I
55: (exp(H)+exp(-H))/(exp(H)-exp(-H))+J
56: JR[1]CW/X+Z[1,2]
57: (exp(H)+exp(-H))/2+Y[2]
58: (exp(I)+exp(-I))/2+Y[1]
59: "cavdone":
60: for Q=1 to 7 by 2
61: for R=1 to 7 by 2
62: 4QR/ππ(MM-QQ)(NN-RR)+U
63: U(cos(Qπ)cos(Mπ)-1)(cos(Rπ)cos(Nπ)-1)+B
64: R[2]T(WW-0[(Q+1)/2,(R+1)/2]+2)/W+Z[2,2]
65: "FIRST MODE PANEL DAMPING":
66: if Q+R>2.5;sto +2
67: R[2]TZ0[(Q+1)/2,(R+1)/2]+2/W+Z[2,1]
68: 1+K[1]
69: 1+K[2]
70: if M=0;.5+K[1]
71: if N=0;.5+K[2]
72: 16/ππQR+K[3]
73: K[1]K[2]K[3]BY[1](-1)+((M+N)/2)/Y[2]+S
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74: (Z[1,1]+Z[2,1])↑2+(Z[1,2]+Z[2,2])↑2÷K
75: P[1]+(Z[1,1](Z[1,1]+Z[2,1])+Z[1,2](Z[1,2]+Z[2,2]))S/K÷P[1]
76: P[2]+(Z[1,2](Z[1,1]+Z[2,1])-Z[1,1](Z[1,2]+Z[2,2]))S/K÷P[2]
77: next R
78: next Q
79: next N
80: next M
81: r(P[1]↑2+P[2]↑2)÷P
82: sfs 14;plt log(F),-20log(P),-2;cf= 14
83: next F
84: pclr:des
85: scl 0,200,-10,50;csiz 1.4,2,.77
86: fxd 0;vax 0,10,-10,50,-1
87: plt -7.3,5.5,1
88: csiz 1.5,1,.77,90
89: wrt 705,"CS4";lbl "INSERTION LOSS → dB"
90: csiz 1.4,2,.77
91: dim D[20];-2.16÷D[1];12.25÷D[2]
92: 22.65÷D[3];30.87÷D[4];37.3÷D[5]
93: 43.08÷D[6];47.8÷D[7];52.1÷D[8]
94: 61÷D[9];95.7÷D[10];105.89÷D[11]
95: 113.88÷D[12];120.62÷D[13];125.96÷D[14]
96: 131.03÷D[15];135.25÷D[16];164÷D[17]
97: 178.6÷D[18];189.09÷D[19];197.1÷D[20]
98: I÷I
99: for N=2 to 9
100: plt D[I],-11.06,1;lbl N
101: I÷I+1
102: if I>20;sto +3
103: next N
104: sto -5
105: plt 58.1,-10,1
106: iplt 0,-1.45,2;plt 55.9,-12.45,1
107: lbl "100";plt 141.5,-10,1
108: iplt 0,-1.45,2;plt 138,-12.45,1
109: lbl "1000";plt 71,-14.4,1
110: csiz 1.5,1,.77
111: lbl "FREQUENCY → Hz"
112: end

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